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THE SOCIETY OF AUTOMOBILE ENGINEERS

1916

TRANSACTIONS

PART II

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THE SOCIETY OF AUTOMOBILE ENGINEERS

1916 TRANSACTIONS

PART II

VOLUME XI

Comprising Papers and Reports and Discussion of Same Presented at the June, 1916, Meeting of the Society, and some Papers Presented at Meetings of Sections of the Society.

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29 W. 39th STREET, NEW YORK

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The
Society of Automobile Engineers

Founded 1905 — Incorporated 1909

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The Society shall not be responsible for statements or opinions advanced in papers or discussions at its meetings * * *

—*Constitution, paragraph* 54.

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SEMI-ANNUAL MEETING

Held on
S. S. Noronic, Detroit-Georgian Bay, June 12-16, 1916

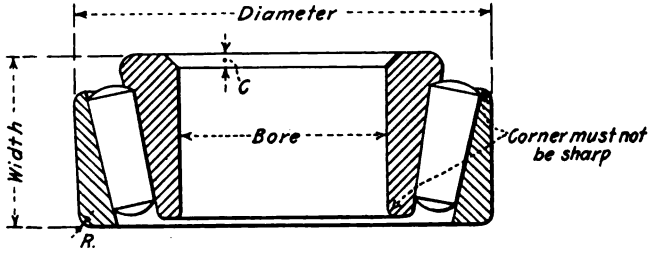
MEETING OF THE STANDARDS COMMITTEE

The Standards Committee meeting was called to order at 3 p. m., June 12, by Chairman Clayden.

SIXTH REPORT OF BALL AND ROLLER BEARINGS DIVISION

(As Adopted by the Society)

A final report is now submitted on dimensions for metric series of roller bearings for both wide and narrow series. These dimensions are given in Tables I and II.



ROLLER BEARING TO ACCOMPANY TABLES I AND II

The Division has found that manufacturers of both straight and taper roller bearings are making the majority of sizes given in the tables. These bearings are now in actual use and being manufactured by various companies.

Respectfully submitted,
BALL AND ROLLER BEARINGS DIVISION,
F. G. HUGHES,
Chairman.

DISCUSSION

C. W. MCKINLEY:—The Division has not met since the April meeting in Cleveland, and we have so far not heard any comments on the proposed taper roller bearing standards submitted at that time. We are working on the end-thrust bearing standardization, and expect to have tables ready to submit within a few months.

(On motion, duly seconded, it was voted that the report of the Ball and Roller Bearings Division be approved for submission to the Society for adoption.)

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TABLE I—METRIC SERIES ROLLER BEARINGS—NARROW SERIES

Bearing Number	Bore				Diameter				Width				Chamfer C Not Less Than		Radius R		Eccentricity Tolerance in Inches	
	Mm.	Inches	Tolerance in Inches		Mm.	Inches	Tolerance in Inches		Mm.	Inches	Tolerance in Inches		Mm.	Inches	Mm.	Inches	Inner Race	Outer Race
			Plus	Minus			Plus	Minus			Plus	Minus						
RM 204.	20	.78740	.0002	.0004	47	1.85040	0	.0006	19.05	3/4	.020	.020	1	.04	1	.04	.0008	.0012
RM 205.	25	.98425	.0002	.0006	52	2.04725	0	.0008	19.05	3/4	.020	.020	1	.04	1	.04	.0008	.0012
RM 206.	30	1.18110	.0002	.0006	62	2.44095	0	.0008	19.05	3/4	.020	.020	1	.04	1	.04	.0008	.0012
RM 207.	35	1.37795	.0002	.0006	72	2.83465	0	.0008	22.22	7/8	.020	.020	1	.04	1	.04	.0008	.0012
RM 208.	40	1.57481	.0002	.0006	80	3.14962	0	.0008	25.40	1	.020	.020	2	.08	2	.08	.0008	.0012
RM 209.	45	1.77166	.0002	.0006	85	3.34647	0	.0008	25.40	1	.020	.020	2	.08	2	.08	.0010	.0016
RM 210.	50	1.96851	.0002	.0006	90	3.54332	0	.0008	25.40	1	.020	.020	2	.08	2	.08	.0010	.0016
RM 211.	55	2.16536	.0002	.0006	100	3.93702	0	.0008	30.16	1 3/16	.020	.020	2	.08	2	.08	.0010	.0016
RM 212.	60	2.36221	.0002	.0006	110	4.33072	0	.0008	34.92	1 3/8	.020	.020	2	.08	2	.08	.0010	.0016
RM 213.	65	2.55906	.0002	.0006	120	4.72443	0	.0008	34.92	1 3/8	.020	.020	2	.08	2	.08	.0010	.0016
RM 214.	70	2.75591	.0002	.0006	125	4.92128	0	.0008	36.51	1 7/16	.020	.020	2	.08	2	.08	.0010	.0016
RM 215.	75	2.95277	.0002	.0006	130	5.11813	0	.0008	36.51	1 7/16	.020	.020	2	.08	2	.08	.0010	.0016
RM 216.	80	3.14962	.0002	.0006	140	5.51183	0	.0008	41.27	1 5/8	.020	.020	3	.12	3	.12	.0012	.0018
RM 217.	85	3.34647	.0002	.0007	150	5.90554	0	.0012	44.45	1 3/4	.020	.020	3	.12	3	.12	.0012	.0018
RM 218.	90	3.54332	.0002	.0007	160	6.29924	0	.0012	50.80	2	.020	.020	3	.12	3	.12	.0012	.0018
RM 219.	95	3.74017	.0002	.0007	170	6.69294	0	.0012	55.56	2 3/16	.020	.020	3	.12	3	.12	.0012	.0018
RM 220.	100	3.93702	.0002	.0007	180	7.08664	0	.0012	60.32	2 3/8	.020	.020	3	.12	3	.12	.0012	.0018
RM 221.	105	4.13387	.0002	.0007	190	7.48035	0	.0012	65.09	2 9/16	.020	.020	3	.12	3	.12	.0012	.0018
RM 222.	110	4.33072	.0002	.0007	200	7.87405	0	.0012	69.85	2 3/4	.020	.020	3	.12	3	.12	.0012	.0018

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TABLE II—METRIC SERIES ROLLER BEARINGS—WIDE SERIES

Bearing Number	Bore				Diameter				Width				Chamfer C Not Less Than		Radius R		Eccentricity Tolerance in Inches	
	Mm.	Inches	Tolerance in Inches		Mm.	Inches	Tolerance in Inches		Mm.	Inches	Tolerance in Inches		Mm.	Inches	Mm.	Inches	Inner Race	Outer Race
			Plus	Minus			Plus	Minus			Plus	Minus						
RM 304.	20	.78740	.0002	.0006	52	2.04725	0	.0008	22.22	7/8	.020	.020	1	.04	1	.04	.0008	.0012
RM 305.	25	.98425	.0002	.0006	62	2.44095	0	.0008	25.40	1	.020	.020	1	.04	1	.04	.0008	.0012
RM 306.	30	1.18110	.0002	.0006	72	2.83465	0	.0008	30.16	1 3/16	.020	.020	2	.08	2	.08	.0008	.0012
RM 307.	35	1.37795	.0002	.0006	80	3.14962	0	.0008	34.92	1 3/8	.020	.020	2	.08	2	.08	.0008	.0012
RM 308.	40	1.57481	.0002	.0006	90	3.54332	0	.0008	36.51	1 7/16	.020	.020	2	.08	2	.08	.0008	.0012
RM 309.	45	1.77166	.0002	.0006	100	3.93702	0	.0008	39.69	1 9/16	.020	.020	2	.08	2	.08	.0010	.0016
RM 310.	50	1.96851	.0002	.0006	110	4.33072	0	.0008	44.45	1 3/4	.020	.020	2	.08	2	.08	.0010	.0016
RM 311.	55	2.16536	.0002	.0006	120	4.72443	0	.0008	49.21	1 15/16	.020	.020	2	.08	2	.08	.0010	.0016
RM 312.	60	2.36221	.0002	.0006	130	5.11813	0	.0008	53.97	2 1/8	.020	.020	2	.08	2	.08	.0010	.0016
RM 313.	65	2.55906	.0002	.0006	140	5.51183	0	.0008	58.74	2 5/16	.020	.020	3	.12	3	.12	.0010	.0016
RM 314.	70	2.75591	.0002	.0007	150	5.90554	0	.0012	63.50	2 1/2	.020	.020	3	.12	3	.12	.0010	.0016
RM 315.	75	2.95277	.0002	.0007	160	6.29924	0	.0012	68.26	2 11/16	.020	.020	3	.12	3	.12	.0010	.0016
RM 316.	80	3.14962	.0002	.0007	170	6.69294	0	.0012	68.26	2 11/16	.020	.020	3	.12	3	.12	.0012	.0018
RM 317.	85	3.34647	.0002	.0007	180	7.08664	0	.0012	73.02	2 7/8	.020	.020	3	.12	3	.12	.0012	.0018
RM 318.	90	3.54332	.0002	.0007	190	7.48055	0	.0012	73.02	2 7/8	.020	.020	3	.12	3	.12	.0012	.0018
RM 319.	95	3.74017	.0002	.0007	200	7.87405	0	.0012	77.79	3 1/16	.020	.020	3	.12	3	.12	.0012	.0018
RM-320.	100	3.93702	.0002	.0007	215	8.46460	0	.0012	82.55	3 1/4	.020	.020	3	.12	3	.12	.0012	.0018

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SEVENTH REPORT OF CARBURETER FITTINGS DIVISION

(As Adopted by the Society)

Final reports are submitted on the following subjects: Carbureter Flanges for Eight- and Twelve-Cylinder Engines; Carbureter Pipe-Tap Sizes for Gasoline Tube Unions; Carbureter Throttle-Lever Rod-End; and Carbureter Throttle-Lever Throw. All of these except the last were approved by the Standards Committee at the meeting in April.

The subject of flexible tubing is awaiting the action of the Electrical Equipment Division on the same subject.

CARBURETER FLANGES FOR 8 AND 12 CYLINDER ENGINES

Two general types of carbureters are involved in this subject, one with a single outlet and the other with two outlets. For the former the Division recommends the use of the present standard flange.

For the two-outlet or duplex type it is not considered advisable to recommend a standard. Where the outlets are parallel it does not seem possible to standardize the center-to-center distance without imposing serious restrictions on carbureter development or without doing injustice to some carbureter maker. Where the outlets may be other than parallel too many angles and other factors are involved for standardization.

CARBURETER PIPE-TAP SIZES FOR GASOLINE TUBE UNIONS

The above caption is suggested instead of "Gasoline Pipe Sizes" for the table given in the S. A. E. Handbook, Volume I, sheet 35a, and instead of "Float Chamber Sizes" recommended in the Fourth Report of the Carbureter Fittings Division, adopted June, 1912, and given in Vol. 7, Part II, page 81 of the Transactions. The change in captions is recommended because it is thought the previous ones used are not sufficiently comprehensive or clear.

It is recommended that this table be revised to include pipe sizes for carbureters as follows: ½ and ⅝-in. sizes, ⅛-in. pipe; 2½-in. size, ¼-in. pipe; and 3 and 3½-in. sizes, ⅜-in. pipe. The new sizes added are necessary to make the table complete, since the list of carbureter flange sizes has been increased.

CARBURETER THROTTLE-LEVER ROD-END

The data on this standard as given in the S. A. E. Handbook, Volume I, sheet 35b, should be revised to read: 3/16-in. rod-end for carbureters from ½ to 1 in. inclusive; ¼-in. rod-end for carbureters from 1¼ to 3½ in. inclusive. This revision is necessary in order to cover the entire range of sizes for which standard flanges have been adopted.

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CARBURETER THROTTLE-LEVER THROW

The Division has had this subject under discussion for almost a year. Various phases of it have been considered from the viewpoints of motor-car, steering-gear and carbureter manufacturers, resulting in the following tabular recommendation:

Carbureter Sizes, In.	Linear Throw, In.
½, ⅝, ¾	1¼
1, 1¼, 1½, 1¾, 2	1¾
2½, 3, 3½	2½

By "Linear Throw" is meant the distance in a straight line between the centers of the throttle-lever end hole at closed and fully opened positions of the throttle. It is obvious that the car manufacturers should provide amply for the "Linear Throw," in order to prevent limitations in some other parts of the connecting linkage from interfering with the function of the carbureter stops.

Respectfully submitted for approval as standard practice.

CARBURETER FITTINGS DIVISION,

JEROME J. AULL,

Chairman.

DISCUSSION

J. J. AULL:—At the Cleveland meeting the Committee approved three recommendations, and the Division now presents a final report on carbureter throttle-lever throw.

HENRY SOUTHER:—I hope that the tubing, valves and the connections back to the tank will be considered. We want to simplify such details as much as possible in the aeronautic service. The interchangeability and standardization of parts from the tank to the carbureter for aero-engines in army service are exceedingly important.

J. J. AULL:—The subject of fittings for aeroplane service has not been assigned to the Division. I had occasion personally to design some gasoline cocks, check-valves and unions for the Curtiss Aeroplane Company, which demanded a light-weight brazed union suitable for connection to a flexible tubing.

(On motion duly seconded, it was voted that the report of the Carbureter Fittings Division be approved for submission to the Society for adoption.)

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NINTH REPORT OF THE ELECTRICAL EQUIPMENT DIVISION

(As Adopted by the Society)

The Division has been engaged on three main subjects: 1. Standard dimensions for flexible tubing now complete and presented to the Society. 2. Standardization of various parts of lamps, of bulbs and of their accessories. Here some recommendations are ready but the bulk of the work remains to be done. 3. Standardization of generator, starting motor and ignition-distributor bases and attachments. This is a difficult subject and there are not yet any recommendations in final form.

The Lamp Standards Sub-division has settled a number of questions and the recommendations thus obtained are offered to the Society. The Sub-division has conducted a number of tests of bulbs and of reflectors by the courtesy of the National Lamp Works in that company's splendidly equipped laboratory.

As a consequence of these tests it appears that it soon will be possible to recommend dimensions for bulb filaments and specifications for reflectors, which will relieve as far as possible the present difficulty of obtaining sufficient illumination without glare.

More tests will have to be made and the work will be continued throughout the summer. So far 46 bulbs and 11 reflectors have been given a preliminary test, enabling certain designs to be selected as the best. More careful photometric tests are to be made and the recommendations when submitted will be scientifically accurate.

Standardization of lighting, starting and ignition units will have to proceed slowly. At present many developments are in progress and it is difficult and even inadvisable to make specific recommendations. It appears likely that the methods of attachment for generators may change considerably by the time the 1918 models are completed, so the Sub-division working on this subject wishes to proceed with proper caution. Incidentally, it appears that the discussions taking place at the meetings are likely to have the effect of obtaining agreement that certain methods are the best from a truly fundamental viewpoint. Thus if the work goes slowly its thoroughness and lasting quality will be enhanced thereby.

FLEXIBLE STEEL TUBING

The sizes of flexible steel tubing in Table I are recommended for automobile electric wiring.

Widths of strip and depth of flange are left optional with the manufacturer, so long as the required degree of flexibility and the specified inside and outside diameters are obtained.

The table relates to packed tubing, but the same outside diameters

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should exist in unpacked tubing. It is considered desirable that packing should be used for the sake of tightness. Cotton packing can be used where a fireproof material is not required.

Where galvanized strip is used, it should be electrically galvanized and the amount of zinc should not be less than 0.06 oz. per sq. ft. of surface.

TEST FOR GALVANIZING

The copper-sulphate method is recommended for testing the galvanizing of flexible tubing. This cannot be considered as an exact test, but it does not require a chemist. With experience, anyone can determine quickly by comparison the amount of zinc coating.

TABLE I—SIZES OF FLEXIBLE STEEL TUBING

Listed Size Inside Diameter, In.	Maximum Outside Diameter, In.	Minimum Thickness of Strip, In.
3/16	17/64	0.010
7/32	19/64	0.010
¼	11/32	0.010
9/32	⅜	0.012
5/16	13/32	0.012
⅜	15/32	0.012
7/16	35/64	0.012
½	39/64	0.014
9/16	11/16	0.014
⅝	¾	0.014
11/16	27/32	0.014
¾	29/32	0.016
13/16	31/32	0.016
⅞	1 1/32	0.016
1	1 5/32	0.016

The test solution is to be prepared by boiling pure water with sufficient crystals of chemically-pure copper sulphate to give a saturated solution with an excess of copper sulphate when cool. The solution shall be neutralized by the addition of oxide of copper, then filtered and diluted to a specific gravity of 1.186 at 65 deg. F. (A saturated solution requires about 27 parts of copper sulphate to 73 parts of water, by weight.)

The solution should not be kept in a room whose temperature falls much below 65 deg. F., in order to guard against changes in specific gravity or saturation.

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Method of Making Test

The solution shall be poured into a No. 1 tall glass beaker about 2 in. diameter; about 100 cu. cm. shall be used for test of ¾ in. and smaller conduits, and about 125 cu. cm. for larger sizes. The same supply of solution shall be used for successive dips of any one sample, but a fresh supply shall be used for each new sample, and the used solution shall not be returned to the supply bottle.

The solution in the beaker shall be brought to a temperature of approximately 65 deg. F. before the test is made, by placing the beaker in a larger vessel containing warmer or colder water and stirring with a glass thermometer.

Pieces of conduit about 6 in. long shall be used for the test, with the ends cut off approximately square, cooled before the test if heated by sawing. Each sample shall be washed in running water, dipped up and down in a vessel containing either carbon tetrachloride or ether, and allowed to dry before being placed in the test beaker.

The sample as thus prepared shall be stood on end in the copper-sulphate solution for one minute, without moving or stirring the solution. At the end of one minute the sample is to be removed, rinsed in running water, and wiped lightly, inside and outside, with white cheese-cloth until dry. Violent rubbing and contact with the hand or any other object are to be avoided.

Interpretation of Test

The zinc coating of the sample is dissolved by the solution and a coating of copper deposited in its place. As long as zinc remains the copper can be easily removed by washing and wiping, but when the zinc has been entirely dissolved at any point the copper deposited at that point will not wash or wipe off. The number of dips required to cause a fixed copper deposit is, therefore, in some degree, a measure of the thickness of the zinc coating.

A sample of flexible tubing shall pass this test if it does not have more than 25 per cent of its immersed surface covered with a fixed copper deposit after one immersion for one minute. Several samples should be tested and a comparison noted.

Each sample shall be subjected to at least one additional dip, and the dips are generally continued until the entire immersed surface is coated with a fixed copper deposit.

HEAD-LAMP SUPPORTS

Three standard sizes are recommended for the forked type of head-lamp support, the forks having center-to-center widths of 7¼, 8¼ and 9¼ in. The upper ends of the supports are to be ½ in. diameter, with ½ in. S. A. E. threads and machined shoulders not less than ¾ in. diameter. The distance from the upper face of the shoulder to the last full thread on the end of the support should not be less than 1½ in. where no tie-rod is used, or 1½ in. plus

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thickness of rod where a rod is used. The use of nuts and lock washers for locking lamp to the fork is recommended as standard practice.

An adjustment should be provided for the support to allow a change of the vertical angle of the lamp without bending any part of the support. The lugs attached to the lamp shells shall have bores of 17/32 in., the bores being 1½ in. long. The center-line of the hole in the lug shall be not less than 9/16 in. from the nearest point of the shell. The clearance between the lower part of the bracket and the lamp shall not be less than 9/16 in.

MOUNTING OF HEAD-LAMPS

It is recommended that head-lamps be mounted as high as practicable, and that the centers of the lamps should never be located less than 3 ft. from the ground. It is found that with a lamp placed high the same intensity of illumination can be obtained with a smaller candlepower than when the lamp is placed low.

Dimming devices that operate by reducing the current to the head-lamp bulbs are not recommended *for the purpose of eliminating glare.*

HEAD-LAMP BULB RECEPTACLES

It is recommended that Sheet 38b, Vol. I of the Handbook, be revised to read: The positions of slots in head-lamp bulb receptacles shall be fixed so that the anchor-pins of the bulbs shall lie in a vertical plane.

FOCUSING DEVICES AND BULB-BASE STANDARDS

It is also recommended that all focusing devices be so arranged as neither to revolve nor cause the bulb to move out of its axis.

It is recommended to delete from the drawings of bulb bases on Sheets 38 and 38xa, Vol. I of the Handbook, the dimensions for gage of metal and diameter of holes for wires.

GROUND-RETURN INSTALLATIONS

It is recommended that the second paragraph on Sheet 48f shall read as follows:

The storage battery should be "grounded" by only one conductor. This "ground" connection should be made readily detachable. The switch for the starting motor should be placed in the ungrounded side of the circuit between the battery and the motor.

Respectfully submitted.

ELECTRICAL EQUIPMENT DIVISION,

A. L. RIKER,
Chairman.

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DISCUSSION

CHAIRMAN CLAYDEN:—The next report is that of the Electrical Equipment Division. Mr. Riker, who unfortunately is not present, has asked me to give the report.

The Division is working on numerous problems connected with the design of lamps. A Sub-division was appointed under the chairmanship of Mr. McKechnie to study the effect of different types of filaments and reflectors. Last Friday at the National Lamp Works at Cleveland we tested four different types of filaments and some fourteen different reflectors. The work has not arrived at a conclusive stage, but we have made a great deal of progress. There was no question at all but that certain designs of filament were in every way superior to certain other designs. Those eliminated first were the extremely fancy patterns. The ordinary filaments seemed to be the most satisfactory from a light-giving point of view. The reflectors certainly showed up better than we expected. The only really weak spot was the focusing arrangements, some of which were not at all efficient. It will be difficult to recommend a standard form of focusing arrangement. Any displacement of the axis from the center of the reflector is detrimental to the efficiency.

The recommendations relating to flexible steel tubing are entirely in accordance with the manufacturing possibilities and with the ideas of the users of tubing among automobile manufacturers.

At the meeting of the Lamp Standards Sub-division of the Electrical Equipment Division held May 24, a number of recommendations were considered and have since been accepted by the Division. The first subject concerns the dimensions of brackets for the head-lamp support. Where these were of the fork type it was found that the center distance between the arms varied in practice from 7 to 10½ in. After considerable discussion it was agreed that three standard sizes, 7¼, 8¼ and 9¼ in., would meet all requirements.

The recommendation previously accepted by the Electrical Equipment Division, that the socket pins on the bulbs should be placed horizontally on the lamps, was withdrawn. This was done because sidewise movement of the lamp from its central position does not affect the distribution of light as seriously as deviation in the vertical plane.

It was resolved that lamps should be mounted as high as possible, and never less than 3 ft. high, measuring from the ground to the center of the lamp. As the lamp is raised above the ground the same intensity of illumination can be obtained with a smaller candlepower. The maximum height advisable has not yet been determined. It is probably in the neighborhood of 4 ft. It was first proposed to recommend that the center of the lamp be 3 ft. 6 in. above the ground; the reason for changing to 3 ft. was that it is difficult to mount lamps any higher on the average chassis.

FLEXIBLE-TUBING DIMENSIONS

J. J. AULL:—There is a slight discrepancy between the present

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recommendation on flexible tubing and the recommendation of the Carbureter Fittings Division a year ago. The air-heater bore was given in figures representing the outside diameter of tubing, and we listed a 1-in. outside diameter. This was the smallest size, and the next was 1⅛ in. We have in this recommendation diameters of 31/32 and 1 1/32 in. Possibly either one can be used in the 1-in. bore adopted for the air-heater for the ½-in. carbureter. The tube enters the air-heater at the exhaust pipe and enters the carbureter. Of course the ½-in. carbureter is not involved in automobile practice, but we ought to be consistent in the two recommendations. If it were possible to add a 1-in. diameter, that would overcome our objection.

H. G. OSBURN:—It would be hard to include that and make a tubing strong enough, because as the tubing increases in size the depth of the flange should increase in order to give permanency to the lock. If we make the wall too thin we weaken the tubing. We might add a size and call it something else besides 1-in. tubing. Possibly listing an 11/16-in. I. D. tubing would obviate Mr. Aull's objection.

J. J. AULL:—No, it would have to be about 25/32-in. I. D. Why do you make the inside diameter the normal size rather than the outside diameter?

H. G. OSBURN:—This has been the practice. The manufacturers to whom we wrote and who use this tubing thought this preferable.

J. J. AULL:—Our recommendation a year ago was based on the outside diameter.

H. G. OSBURN:—From the electric viewpoint it seems desirable to keep fairly close to the inside diameter. If we could include a few more sizes, that would perhaps take care of the carbureter recommendation.

J. J. AULL:—Rather than introduce a new size I think it would be better to adopt the sizes the Electrical Equipment Division has listed, because the change involves only ½ and ⅝-in. carbureters. Above that we can carry out our present list of sizes.

HEIGHT OF LAMPS

C. W. MCKINLEY:—The recommendation as to height of lamps would make some cars look like frogs. The lamp would be above the radiator.

CHAIRMAN CLAYDEN:—It was realized that the height recommended could not always be used, but we worked the subject out carefully. A lamp will certainly give more light with a high than with a low location.

HENRY SOUTHER:—About six items in the list of subjects before the Electrical Equipment Division should, it seems to me, be handled in relation to possible future use in the air. The safety of wiring devices in the air is of great importance. A switch that will not draw a naked spark is also important.

(On motion, duly seconded, it was voted to approve the report of

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the Electrical Equipment Division for submission to the Society for adoption.)

UNDERWRITERS INTERESTED IN AUTOMOBILES

HENRY SOUTHER:—Do the underwriters have any control over the wiring of either gas vehicles or electric vehicles? From the safety standpoint it would be well to get them interested.

H. G. OSBURN:—The underwriters' laboratory makes tests, which, if approved, are recognized by the various insurance companies. The laboratory also has a label service in which goods are inspected at the factory and labeled so that they can be recognized as meeting the approval of the underwriters. I do not know that anything has been put into force as to the finished vehicle receiving inspection, or as to the necessity of this.

The specifications now required by the underwriters in the wiring of buildings include heavier material generally than has been considered necessary for automobile wiring. I am speaking of wiring for ignition, starting and lighting systems in which the voltage is low. This has not, I suppose, been considered as coming within the jurisdiction of the underwriters, but more in the class of telephone and electrical appliances, such as bell and signal circuits, requiring low voltages.

It seems to me that it would be in keeping with our general work to appoint a committee to confer with the underwriters.

CHAIRMAN CLAYDEN:—The work could be done by a joint committee of the Electrical Equipment and Electric Vehicle Divisions.

J. J. AULL:—The present underwriters' rules do not permit the use of a stop-cock to control gasoline, and we know that practically every gasoline automobile made has a stop-cock in the line. Neither will they allow the gasoline supply to be above the carbureter. The rule regarding tractors, which I believe will apply to automobiles also, is to the effect that the receptacle feeding the carbureter shall not contain more than a pint. The supply is to be pumped to this receptacle and returned through an overflow to the main tank, a construction which is impractical for the automobile.

H. G. OSBURN:—The problem affects to a certain extent the insurance in adjoining buildings. We must also guard against placing too great a burden on the industry in the way of inspection fees. The policy of the underwriters has been to cooperate with the manufacturers, holding joint meetings with them. They are not inclined to insist on rules the manufacturers consider impractical; so that in general the details can be worked out, due consideration being given to the requirements on each side.

J. J. AULL:—The underwriters refuse to pass the flared union, the type of the S. A. E. standard, while they will accept the pipe union and the compression union. The compression union is not so good in

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any way as the S. A. E. They will accept the S. A. E. for oil lines but not for gasoline.

(On motion, duly seconded, it was voted that a joint committee of the Electrical Equipment and Electric Vehicle Divisions should confer with the underwriters regarding automobile construction.)

FOURTH REPORT OF ELECTRIC VEHICLE DIVISION

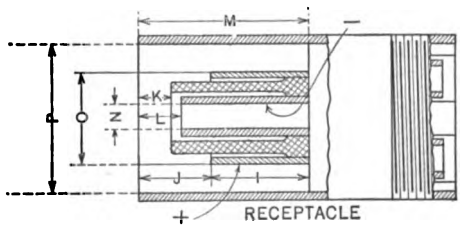
(As Adopted by the Society)

The Electric Vehicle Division has one subject ready for final report. Substantial progress has been made on two other subjects before the Division.

CHARGING RECEPTACLES

It is proposed to revise some of the dimensions of the E. V. A. A. standard charging receptacle shown on Sheet 50, Vol. I, of the Hand-

TABLE I—CHANGES IN CHARGING RECEPTACLE DIMENSIONS



Dimension	50 amp.		150 amp.	
	Proposed	Present	Proposed	Present
J	1⅛	⅞	2	1 5/16
K	⅝	⅜	1	5/16
L	⅝	⅜	1⅛	7/16
M	2¼	2	3 13/32	2 3/32

All dimensions in inches.

book. The receptacle is shown in the accompanying sketch, while Table I gives these dimensions and the proposed changes.

Jars for Lead Batteries

Considerable work has been done in an effort to standardize sizes of jars for lead batteries. Virtual agreement has been reached in all dimensions except the length, which is a variable. A carefully prepared table containing a proposal for the reduction of the numbers

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of lengths in use has been submitted to battery makers, but no agreement has yet been reached. About fifty lengths have been in common use, and it is hoped to reduce the number to about twenty-five or thirty.

Charging Cable

A proposed standardization of charging cable has been under advisement for some time. The Division is now awaiting action of the American Institute of Electrical Engineers in regard to stranding.

Respectfully submitted.

ELECTRICAL VEHICLE DIVISION,
ARTHUR J. SLADE,
Chairman.

DISCUSSION

A. J. SLADE:—The Electric Vehicle Division has one subject ready for report. It is proposed to revise some dimensions of the Electric Vehicle Association standard charging receptacle. The changes involve the lengthening of the shell, in order to obviate the present tendency toward breakage of the shell when the plug is inserted or withdrawn. They have already been put into practice by one of the principal manufacturers of charging plugs and receptacles.

(On motion, duly seconded, it was voted to approve the report of the Electric Vehicle Division for submission to the Society for adoption.)

NINTH REPORT OF THE IRON AND STEEL DIVISION

(*As Adopted by the Society*)

Final reports are now submitted covering (1) physical-property data for steels Nos. 2320 to 3140; (2) the addition of a 3½ per cent nickel steel to be known as No. 2345; (3) the revisions of the sheets in the Handbook on which flat-tension and gray-iron test-specimens are shown. The Division recommends that the subject of standard colors for steels be dropped. A progress report is submitted on the scheme for reclassifying sheet steel.

PHYSICAL PROPERTIES OF ALLOY STEELS

It is recommended that the accompanying tabulations and curves of the physical-property data for steels Nos. 2320 to 3140 be published as supplements to the revised Notes and Instructions in the Eighth Report of the Division. The figures on which these data are based have been derived from various sources, and are to be interpreted according to the system explained in the Seventh Report of the Iron and Steel Division, and as abstracted on Sheet 9g, Vol. I of the Handbook.

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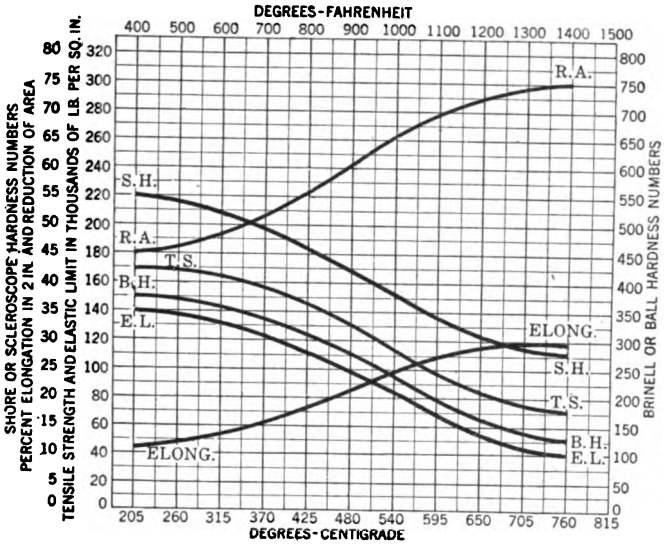
Physical Characteristics of Heat Treated S. A. E. Steel No. 2320



Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, which were heated from 15 to 30 min. at 1510 to 1540 deg. F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air.

Tabulation of Values Plotted in Curves*

Reheating Temperature, Deg. F.	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Reduction of Area, per Cent	Elongation in 2 In., per Cent	Brinell Hardness	Scleroscope Hardness
400	170,000	140,000	45.0	11.0	375	55
500	168,000	136,500	46.0	12.0	368	54
600	162,000	130,000	48.5	13.5	355	52
700	155,000	123,000	51.5	15.5	340	50
800	145,000	112,000	55.5	18.5	310	46
900	130,000	99,000	60.5	21.5	280	42
1000	112,000	84,000	65.5	25.0	240	38
1100	96,000	68,000	69.5	27.0	200	34
1200	82,000	54,000	72.5	29.0	165	31
1300	75,000	45,000	74.5	29.5	140	29
1400	70,000	40,000	75.0	30.0	125	28

*Values are average minimum, except those for hardness, which are average.

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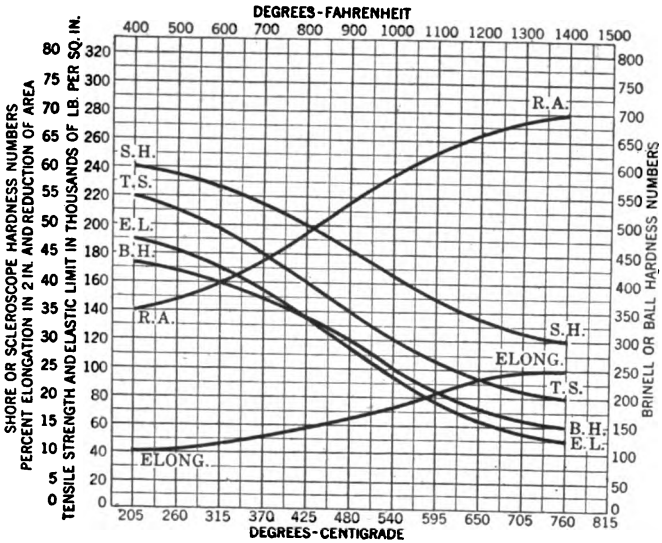
Physical Characteristics of Heat Treated S. A. E. Steel No. 2330



Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, which were heated from 15 to 30 min. at 1485 to 1515 deg. F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air.

Tabulation of Values Plotted in Curves*

Reheating Temperature, Deg. F.	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Reduction of Area, per Cent	Elongation in 2 In., per Cent	Brinell Hardness	Scleroscope Hardness
400	220,000	190,000	35.0	10.0	436	60
500	210,000	182,000	37.0	10.7	420	59
600	198,000	170,000	40.0	11.5	400	57
700	180,000	154,000	44.0	12.7	375	54
800	160,000	135,000	49.0	14.5	340	50
900	140,000	115,000	54.0	16.0	300	46
1000	120,000	95,000	59.0	18.0	250	41
1100	104,000	77,000	62.7	20.5	210	37
1200	92,000	64,000	66.0	22.5	180	34
1300	85,000	55,000	68.5	24.5	162	32
1400	80,000	50,000	70.0	25.0	150	30

*Values are average minimum, except those for hardness, which are average.

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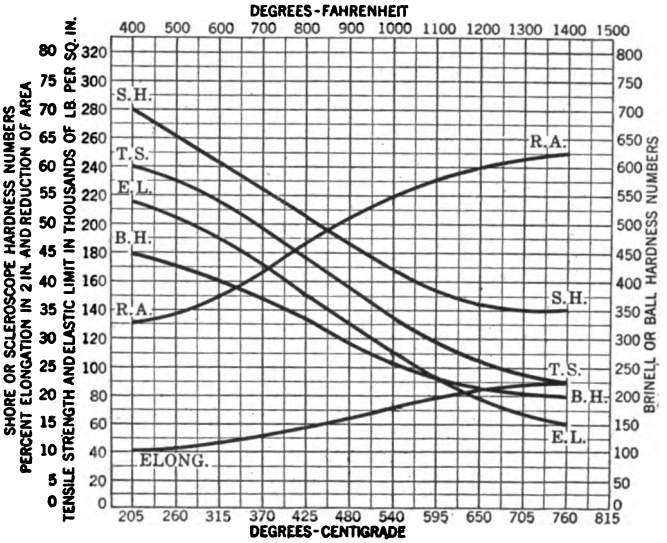
Physical Characteristics of Heat Treated S. A. E. Steel No. 2340



Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, which were heated from 15 to 30 min. at 1435 to 1465 deg. F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air.

Tabulation of Values Plotted in Curves*

Reheating Temperature, Deg. F.	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Reduction of Area, per Cent	Elongation in 2 In., per Cent	Brinell Hardness	Scleroscope Hardness
400	240,000	215,000	32.5	10.0	450	70
500	230,000	204,000	34.5	11.0	427	65
600	215,000	190,000	37.5	12.0	400	61
700	196,000	171,000	42.0	13.0	370	56
800	175,000	150,000	47.0	14.0	335	51
900	155,000	130,000	51.0	16.0	295	46
1000	135,000	110,000	55.0	18.0	260	42
1100	117,000	92,000	58.0	20.0	235	38
1200	105,000	78,000	60.0	21.5	215	36
1300	96,000	69,000	61.0	22.0	205	35
1400	90,000	60,000	62.5	22.5	200	35

*Values are average minimum, except those for hardness, which are average.

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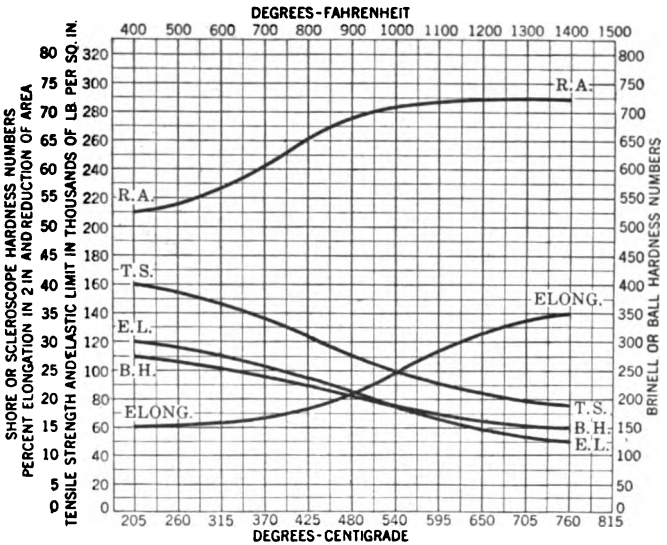
Physical Characteristics of Heat Treated S. A. E. Steel No. 3120



Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, which were heated from 15 to 30 min. at 1585 to 1615 deg. F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air.

Tabulation of Values Plotted in Curves*

Reheating Temperature, Deg. F.	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Reduction of Area, per Cent	Elongation in 2 In., per Cent	Brinell Hardness	Scleroscope Hardness
400	160,000	120,000	52.5	15.0	275	.....
500	155,000	116,000	54.0	15.5	265	.....
600	148,000	110,000	57.0	16.0	250	.....
700	137,000	102,000	61.0	16.5	240	.....
800	125,000	95,000	65.5	18.0	225	.....
900	111,000	84,000	69.0	21.0	205	.....
1000	100,000	74,000	71.0	24.5	185	.....
1100	91,000	66,000	71.5	28.5	175	.....
1200	84,000	60,000	72.0	31.5	160	.....
1300	80,000	54,000	72.5	33.5	150	.....
1400	75,000	50,000	72.5	35.0	150	.....

*Values are average minimum, except those for hardness, which are average.

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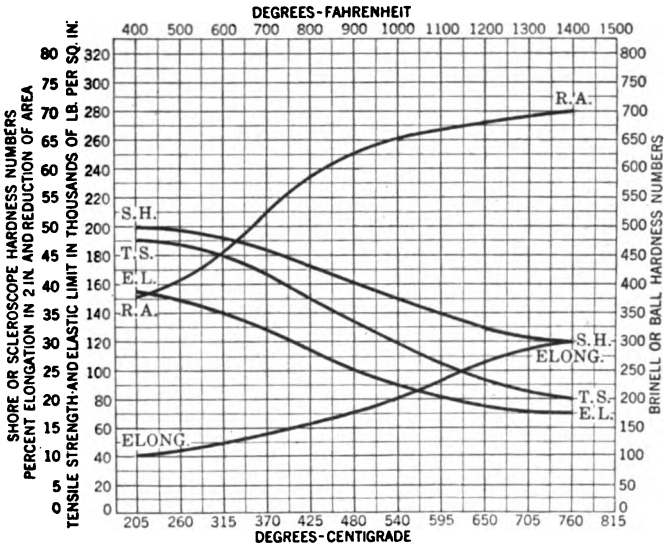
Physical Characteristics of Heat Treated S. A. E. Steel No. 3130



Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, which were heated from 15 to 30 min. at 1535 to 1565 deg. F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air.

Tabulation of Values Plotted in Curves*

Reheating Temperature, Deg. F.	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Reduction of Area, per Cent	Elongation in 2 In., per Cent	Brinell Hardness	Scleroscope Hardness
400	190,000	155,000	37.5	10.0	.....	50
500	188,000	150,000	41.0	11.0	.....	49
600	180,000	140,000	46.0	12.5	.....	48
700	167,000	128,000	52.5	13.5	.....	46
800	150,000	115,000	59.0	15.5	.....	43
900	134,000	102,000	63.0	17.5	.....	40
1000	120,000	90,000	65.0	20.0	.....	38
1100	104,000	81,000	66.5	23.5	.....	35
1200	92,000	76,000	68.0	26.5	.....	32
1300	86,000	72,000	69.0	28.5	.....	31
1400	80,000	70,000	70.0	30.0	.....	30

*Values are average minimum, except those for hardness, which are average.

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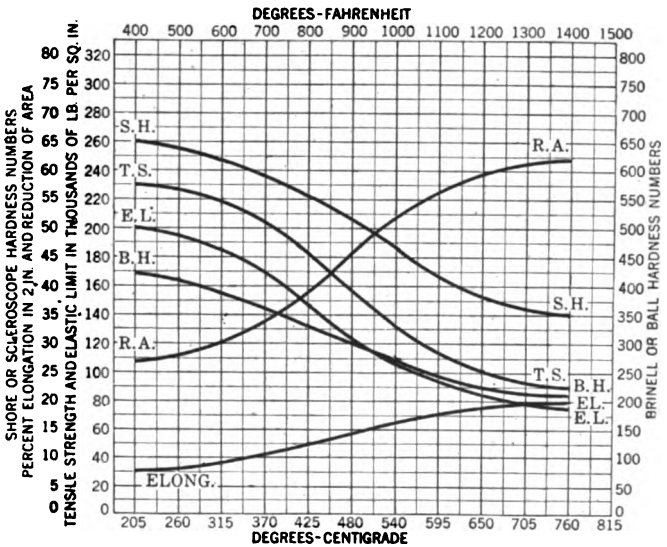
Physical Characteristics of Heat Treated S. A. E. Steel No. 3140



Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, which were heated from 15 to 30 min. at 1485 to 1515 deg. F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air.

Tabulation of Values Plotted in Curves*

Reheating Temperature, Deg. F.	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Reduction of Area, per Cent	Elongation in 2 In., per Cent	Brinell Hardness	Scleroscope Hardness
400	230,000	200,000	27.0	7.5	425	65
500	226,000	195,000	28.0	8.0	410	64
600	220,000	185,000	30.0	9.0	390	62
700	204,000	168,000	34.0	10.5	360	59
800	182,000	148,000	39.0	12.5	330	56
900	157,000	126,000	46.5	14.0	300	52
1000	130,000	105,000	52.5	16.0	275	47
1100	112,000	94,000	56.5	17.0	245	42
1200	100,000	84,000	60.0	18.0	225	38
1300	93,000	80,000	61.0	19.0	215	36
1400	90,000	75,000	62.0	20.0	210	35

*Values are average minimum, except those for hardness, which are average.

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HIGH-CARBON NICKEL STEEL

It is recommended that a 3½ per cent nickel steel containing carbon from 0.40 to 0.50 per cent and with the other elements the same as in steel No. 2340 be added to the list of standard analyses. This steel is rather widely used by a number of companies for oil-treated gears, and appropriate remarks concerning it will be included in the next full reprint of the Notes and Instructions.

FINISH ON FLAT TEST-SPECIMEN

It is recommended that a note be added to the data sheets on standard and alternative flat-tension test-specimens as follows:

"The gage length shall be finished on all sides smooth and free from scratches." The same note should be included in the text accompanying sketches of these specimens wherever they appear in specifications.

GRAY IRON TEST-SPECIMENS

It is recommended that the gray iron tension test-specimen be removed from the present position (Sheet 15c, Vol. I of Handbook) and put on a separate sheet. A note is to be added as follows: "The gage length shall be ground or lapped smooth and free from scratches."

Arbitration Bar Mold

It is recommended that the following note be added to a sheet headed Arbitration Bar Mold: "This pattern and mold (see sheet 15c) are to be used in making castings for the production of standard transverse and tension test-specimens of gray cast-iron. The conditions for molding and casting are given in the following paragraphs."

METHOD OF CASTING ARBITRATION BARS

It is recommended that a data sheet entitled, Method of Casting Arbitration Bars for Gray Iron Tests, be printed containing the following:

A number of bars shall be cast from each heat sufficient to be thoroughly representative of the heat for the purpose of the tests. Each set of two bars is to go into a single mold, and the resulting bars shall not be rumbled or otherwise treated, being simply brushed off before testing.

The pattern for the bars is shown on Sheet 15c, Vol. I of the Handbook, and is 1/16 in. smaller in diameter at the bottom than at the top to allow for draft and for the strain of pouring. The pattern shall not be rapped before being withdrawn. The flask is to be rammed up with green molding sand, a little damper than usual, well mixed and put through a No. 8 sieve, with a mixture of one to twelve bituminous facing. The mold shall be rammed evenly and fairly hard, thoroughly dried and not cast until it is cold. The test bar shall not be removed from the mold until cold enough to be handled.

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Transverse tests shall be made on all the bars cast, with supports 12 in. apart, load applied at the middle and the deflection at rupture noted. Of each set of two bars, one must fulfill the specified requirements to permit acceptance of the castings represented. The rate of application of the transverse load shall be from 20 to 40 sec. for a deflection of 0.10 in.

Tensile tests are not recommended, but in case they are called for, the test-specimens can be turned from the broken pieces left from the transverse test.

Standard Colors for Steels

It is recommended that the subject of standard colors for steels be dropped, and that the Sub-division on the subject be discharged. Although the present situation on this subject is chaotic, no two firms using the same colors for the same steel throughout, there are many reasons why the Committee cannot hope to establish a standard scale. Among these are the small number of colors that remain permanent in the weather; the large number of steels in our list; and the consequent necessity of using combinations of two, three or even more colors for many of the steels.

Automobile Sheet Steel

Progress is reported on the tentative scheme for reclassifying sheet steel according to temper and finish. This is a large subject and will require a long time for completion, but we are receiving encouragement from the manufacturers and users of sheet in the work.

This report has been submitted to the twelve members of the Division, eight of whom have voted affirmatively, none negatively, and four have refrained from voting for adoption.

Respectfully submitted.

IRON AND STEEL DIVISION,
K. W. ZIMMERSCHIED,
Chairman.

DISCUSSION

K. W. ZIMMERSCHIED:—Since the Cleveland meeting of the Standards Committee, the matters before the Iron and Steel Division have been embodied in a formal report concerning which there are but few remarks to be made.

The American Society for Testing Materials is working on a revision of its standard 2-in. tension test-specimen. We will probably have to revise our standard also, to keep as great concordance between the Societies as possible. The change consists only of eliminating all specifications as to the ends of the test-specimen beyond the fillets that terminate the gage length. The attachment shoulders are to be optional in size and shape, the only specifications being that the line of stress must be coincident with the principal axis of the specimen.

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BRITISH STEEL SPECIFICATIONS

The Engineering Standards Committee of Great Britain issued lately a set of standards for automobile steels. These are quite different, in spirit and in extent, from ours. We have not gone over them in detail, and, not knowing just what considerations moved the Committee, cannot criticize the standards intelligently. We are glad this Committee has finally adopted such standards, because they constitute another common interest between foreign automobile manufacturers and us. We hope that this may be one of the points of mutual interest on which we can get together some day for the formulation of international standards, insofar as they may be desirable for automobile manufacturers.

Just how far such standardization may be advisable is a problem; the International Society for Testing Materials has considered this matter for many years, and there are many reasons why international standardization of steel materials has not progressed rapidly. We may find that the automobile industry is different in this respect and would support greater unanimity. Our respective national standards should then assist us in any program leading to final international standards.

(On motion, duly seconded, it was voted that the report of the Iron and Steel Division be approved for submission to the Society for adoption.)

EIGHTH REPORT OF THE MISCELLANEOUS DIVISION

(As Adopted by the Society)

1. Final reports are now submitted on three subjects, together with progress reports on other subjects. The recommendation in regard to gearshift positions was approved by the Standards Committee at its April meeting, while the matters relating to piston-ring grooves and increased length of thread of S. A. E. bolts are now submitted for approval.

GEARSHIFT POSITIONS

2. It is recommended that Sheets 46, 46a and 46b, Vol. I of the Handbook, showing gearshift gates for both three-speed and four-speed transmissions, be superseded by a new sheet showing the gearshift positions, as in Fig. 1, thus eliminating any reference to positions for four-speed transmissions. The proposed standard relates to the positions regardless of whether a gate is used.

PISTON-RING GROOVES

3. The Division recommends as standard practice the dimensions given in Table I. The dimensions for maximum diameter at bottom of ring groove, minimum width of groove and maximum width of

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ring are the preferred dimensions and should be retained in case closer limits are used.

4. The depths of groove have been determined by the following formula:

$$G = \sqrt{\frac{0.01D^2}{8} + 0.005}$$

in which G equals depth of groove and D the nominal diameter of piston. The curve in Fig. 2 shows the relation between piston diam-

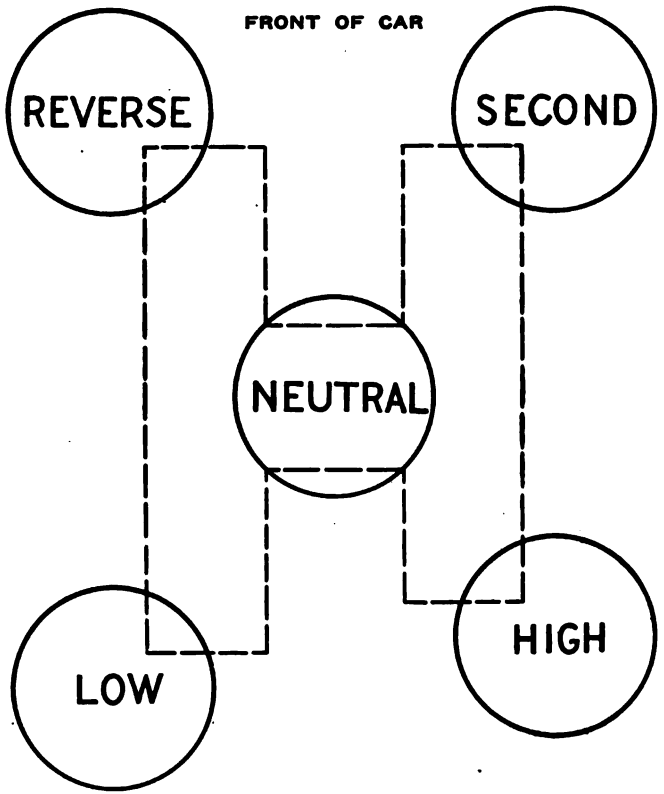

FIG. 1—LAYOUT OF GEARSHIFT POSITIONS

eter and groove depth as determined by this formula. Comments received from the various manufacturers to whom the proposed dimensions for piston-ring grooves were submitted have been considered in detail. Nearly all the replies were favorable to the proposal as outlined.

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TABLE I—PISTON-RING GROOVE WIDTHS AND DEPTHS

Dia. (D)	Depth (G)	Dia. and Limits Bottom Ring Groove		Groove Width	Ring Width
			*	†	*
1	.0791	.8398	.834- .840	.1250-.1260	.1237-.1247
1⅛	.0811	.9606	.955- .961	.1250-.1260	.1237-.1247
1¼	.0834	1.0807	1.075-1.081	.1250-.1260	.1237-.1247
1⅜	.0858	1.2007	1.195-1.201	.1250-.1260	.1237-.1247
1½	.0884	1.3202	1.315-1.321	.1250-.1260	.1237-.1247
1⅝	.0911	1.4396	1.434-1.440	.1250-.1260	.1237-.1247
1¾	.0939	1.5587	1.553-1.559	.1250-.1260	.1237-.1247
1⅞	.0969	1.6775	1.672-1.678	.1250-.1260	.1237-.1247
2	.1000	1.7960	1.790-1.796	.1250-.1260	.1237-.1247
2⅛	.1032	1.9143	1.909-1.915	.1250-.1260	.1237-.1247
2¼	.1064	2.0347	2.029-2.035	.1250-.1260	.1237-.1247
2⅜	.1098	2.1506	2.145-2.151	.1250-.1260	.1237-.1247
2½	.1132	2.2686	2.263-2.269	.1250-.1260	.1237-.1247
2⅝	.1167	2.3864	2.381-2.387	.1875-.1885	.1862-.1872
2¾	.1202	2.5041	2.499-2.505	.1875-.1885	.1862-.1872
2⅞	.1238	2.6216	2.616-2.622	.1875-.1885	.1862-.1872
3	.1275	2.7390	2.733-2.739	.1875-.1885	.1862-.1872
3⅛	.1312	2.8563	2.851-2.857	.1875-.1885	.1862-.1872
3¼	.1349	2.9737	2.968-2.974	.1875-.1885	.1862-.1872
3⅜	.1387	3.0908	3.085-3.091	.1875-.1885	.1862-.1872
3½	.1425	3.2080	3.202-3.208	.1875-.1885	.1862-.1872
3⅝	.1464	3.3250	3.319-3.325	.1875-.1885	.1862-.1872
3¾	.1503	3.4419	3.436-3.442	.1875-.1885	.1862-.1872
3⅞	.1542	3.5588	3.553-3.559	.1875-.1885	.1862-.1872
4	.1581	3.6758	3.670-3.676	.1875-.1885	.1862-.1872
4⅛	.1621	3.7925	3.787-3.793	.2500-.2510	.2487-.2497
4¼	.1661	3.9093	3.904-3.910	.2500-.2510	.2487-.2497
4⅜	.1701	4.0262	4.021-4.027	.2500-.2510	.2487-.2497
4½	.1741	4.1428	4.137-4.143	.2500-.2510	.2487-.2497
4⅝	.1782	4.2593	4.254-4.260	.2500-.2510	.2487-.2497
4¾	.1822	4.3761	4.371-4.377	.2500-.2510	.2487-.2497
4⅞	.1863	4.4926	4.487-4.493	.2500-.2510	.2487-.2497
5	.1904	4.6092	4.604-4.610	.2500-.2510	.2487-.2497
5⅛	.1945	4.7258	4.720-4.726	.3125-.3140	.3110-.3120
5¼	.1986	4.8423	4.837-4.843	.3125-.3140	.3110-.3120
5⅜	.2028	4.9587	4.953-4.959	.3125-.3140	.3110-.3120
5½	.2069	5.0752	5.070-5.076	.3125-.3140	.3110-.3120
5⅝	.2111	5.1816	5.176-5.182	.3125-.3140	.3110-.3120
5¾	.2152	5.3081	5.303-5.309	.3125-.3140	.3110-.3120
5⅞	.2194	5.4245	5.419-5.425	.3125-.3140	.3110-.3120
6	.2236	5.5408	5.535-5.541	.3125-.3140	.3110-.3120
6⅛	.2278	5.6572	5.652-5.658	.3750-.3770	.3735-.3745
6¼	.2320	5.7735	5.768-5.774	.3750-.3770	.3735-.3745
6⅜	.2362	5.8899	5.884-5.890	.3750-.3770	.3735-.3745
6½	.2404	6.0062	6.001-6.007	.3750-.3770	.3735-.3745
6⅝	.2447	6.1224	6.117-6.123	.3750-.3770	.3735-.3745
6¾	.2489	6.2387	6.233-6.239	.3750-.3770	.3735-.3745
6⅞	.2531	6.3551	6.350-6.356	.3750-.3770	.3735-.3745
7	.2574	6.4712	6.466-6.472	.3750-.3770	.3735-.3745

*Preferred maximum dimensions. †Preferred minimum dimensions.

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INCREASED LENGTH OF THREAD ON S. A. E. BOLTS

5. It is recommended that the formula for length of thread on S. A. E. bolts, appearing on Sheet 4, Vol. I of the Handbook, should be revised to read as follows:

$$D \times 1.5 + \tfrac{1}{4} = \text{length (in inches) of effective thread}$$

The length of thread as now specified is not considered sufficient in cases where the parts bolted together are not machined with limits of less than 1/16 in. It is also reported that screw makers have been inclined to measure the thread to the extreme point marked by the die rather than to the point where the thread is full.

Speedometer-Drive Shaft-Ends

6. Constantly changing conditions and new developments are such that it is deemed inadvisable to formulate a definite standard of

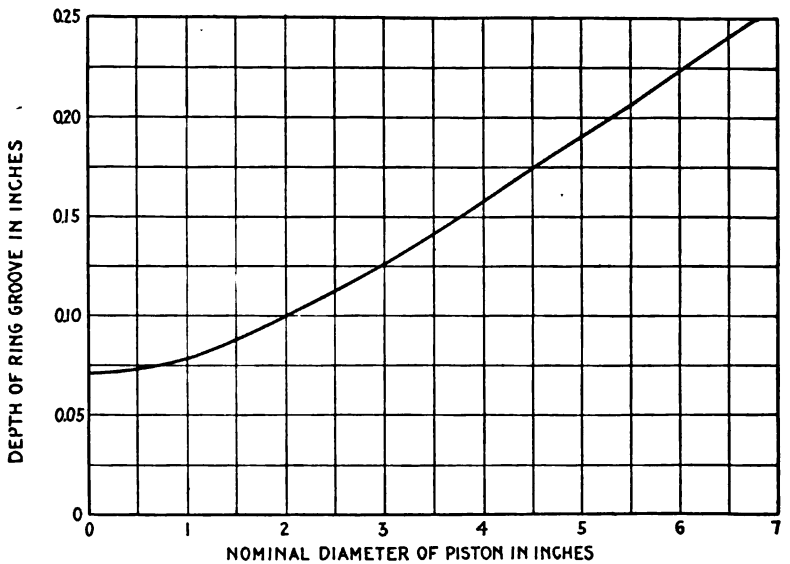

FIG. 2—CURVE SHOWING RELATION OF PISTON DIAMETER AND GROOVE DEPTH

speedometer-drive shaft-ends at this time. Certain phases of the situation indicate that in perhaps six months the subject will be in a better condition for standardization. This is a critical time in the speedometer-drive situation, the whole subject being in process of revolution and evolution; new and improved designs are being presented almost daily and the Division fears that a standard adopted at this time would either limit development of invention or be ignored.

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Taper Sockets for Fender Supports

7. With data received from manufacturers of such material as a basis, a table of dimensions has been prepared for taper fittings for brackets and supports. A taper of 4½ in. per foot is suggested. There are four sizes in the table, with small diameters ranging from 7/16 to ¾ in., and lengths of taper portion from ¾ to 1 3/16 in. This table is to be circulated for comment to manufacturers of top irons, fenders and cars.

Other Subjects

8. The following additional subjects are now before the Division: Steering-wheel hub dimensions; rod-end standards (extension); bolts with various heads; A. S. M. E. pitches for small machine screws; thread tolerances; standardization of drawings; spark-plug terminal thread; and electric horn mounting.

Respectfully submitted.

MISCELLANEOUS DIVISION,

JOHN G. UTZ,

Chairman.

DISCUSSION

J. G. UTZ:—The eighth report is now submitted for approval. The Division has met once since the April Standards Committee meeting in Cleveland. After full returns from the latest circularization on speedometer-drive shaft-ends it was decided to table the subject for the present. We have considerable data on hand on steering-wheel hub dimensions and will be able to make some progress at our next Division meeting. The matters of the A. S. M. E. pitches for small machine screws and the thread tolerances are being held in abeyance until the ideas of the A. S. M. E. on the same subjects are reported.

HENRY SOUTHER:—A new phase of speedometer-drive shaft-ends has developed in the aeroplane, and that is the necessity for transmitting the motion a considerable distance longitudinally. It has been found that the flexible drive is heavy. An aluminum tube and shaft to do the straight driving have been suggested. I think that should be considered at the proper time.

Cable-end standards should now come into common use. There is a variety of opinion now, but a good deal of careful investigation has been made as to the proper method for making an eye-splice in the cable. This is an interesting point and will I hope receive the attention of the Division.

(On motion, duly seconded, it was voted that the report of the Miscellaneous Division be approved for submission to the Society for adoption.)

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REPORT OF THE NOMENCLATURE DIVISION

(As Adopted by the Society)

After a number of meetings held during the past year in conference with engineers and representatives of service departments responsible for the naming of car parts in their own organizations, the following list of names has been prepared and is offered for acceptance by the Society.

An attempt has been made to include in the list the more important parts throughout the whole car, bolts, studs and the like being indicated in general terms. Body parts have not been included generally nor parts of such units as the carbureter, which vary so much in construction as to make anything like uniform nomenclature very difficult.

Definitions of different types of construction have been included for several units in order to encourage uniform terminology in descriptions appearing in the trade press and in catalogs, as well as in the technical discussions of the Society. Definitions of different types of body are also included, because it is thought that some authority should take action to make possible the use of names which will be understood generally, rather than those which are meaningless except to persons conversant with the terminology peculiar to individual manufacturers. It is surprising how many distinctly different types of body are being sold under the name "brougham," for instance.

A scheme of classification based entirely on assemblies is impracticable for general use, on account of diverse arrangement of elements of so-called conventional cars. The classification proposed is therefore based largely on function.

In its present form the report can be used as a guide in making up lists of parts, or for any purpose by those familiar with automobile construction. It is probable that at some time in the near future sketches illustrating various of the parts listed in the report will be supplied.

GENERAL DIVISIONS

I Cylinders
II Valves
III Cooling System
IV Fuel System
V Exhaust System

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VI Lubrication
VII Ignition
VIII Starting and Lighting Equipment
IX Miscellaneous Electrical Equipment
X Clutch
XI Transmission
XII Rear Axle
XIII Braking System
XIV Front Axle and Steering
XV Wheels
XVI Frame and Springs
XVII Hoods, Fenders and Shields
XVIII Body and Top
XIX Accessories

DIVISION I—CYLINDERS

Group 1—Cylinders
Group 2—Crankcase
Group 3—Crankshaft
Group 4—Starting-crank
Group 5—Connecting-rods
Group 6—Pistons

DIVISION II—VALVES

Group 1—Camshaft
Group 2—Valves

DIVISION III—COOLING SYSTEM

Group 1—Fan
Group 2—Radiator
Group 3—Pump
Group 4—Pipes and Hose

DIVISION IV—FUEL SYSTEM

Group 1—Carbureter and Inlet Pipe
Group 2—Carbureter Control
Group 3—Carbureter Air-heater
Group 4—Fuel Tank
Group 5—Fuel Pipes and Feed System

DIVISION V—EXHAUST SYSTEM

Group 1—Exhaust Manifold
Group 2—Exhaust Pipe and Muffler

DIVISION VI—LUBRICATION SYSTEM

Group 1—Oil-pan or Reservoir
Group 2—Oil Pump
Group 3—Oil Pipes, Strainers, Gages

DIVISION VII—IGNITION

Group 1—Spark-plugs, Cables and Switches
Group 2—Ignition Distributor
Group 3—Magneto
Group 4—Ignition Control

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DIVISION VIII—STARTING AND LIGHTING EQUIPMENT

Group 1—Generator
Group 2—Starting Motor
Group 3—Wiring
Group 4—Battery

DIVISION IX—MISCELLANEOUS ELECTRICAL EQUIPMENT

Group 1—Lamps and Wiring
Group 2—Switches and Instruments
Group 3—Horn
Group 4—Miscellaneous

DIVISION X—CLUTCH

Group 1—Clutching Parts
Cone Clutch
Disk Clutch
Plate Clutch
Group 2—Releasing Parts

DIVISION XI—TRANSMISSION

Group 1—Transmission
Group 2—Shifting Mechanism
Group 3—Control
Group 4—Propeller-shaft

DIVISION XII—REAR AXLE

Group 1—Housing
Group 2—Torque-arm and Radius-rod
Group 3—Drive Pinion
Group 4—Differential
Group 5—Axle Shafts

DIVISION XIII—BRAKES

Group 1—Outer Brake
Group 2—Inner Brake
Group 3—Pedal (or outer) Brake Control
Group 4—Hand (or inner) Brake Control

DIVISION XIV—FRONT AXLE AND STEERING

Group 1—Axle Center
Group 2—Steering-knuckles
Group 3—Steering-rods
Group 4—Steering-gear

DIVISION XV—WHEELS

Group 1—Front Wheels
Group 2—Rear Wheels

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DIVISION XVI—FRAME AND SPRINGS

Group 1—Frame
Group 2—Frame Brackets and Sockets
Group 3—Front Springs
Group 4—Rear Springs

DIVISION XVII—HOOD, FENDERS AND SHIELDS

Group 1—Hood
Group 2—Engine Shield
Group 3—Fenders and Running-boards
Group 4—Windshield

DIVISION XVIII—BODY

Group 1—Floor-boards and Dash
Group 2—Body
Group 3—Upholstering
Group 4—Top

DIVISION XIX—ACCESSORIES

Group 1—Speedometer
Group 2—Tire-pump

GENERAL

Where terms "front" and "rear" are used, "front" should always be toward the front end of the car. These terms are sometimes confused in regard to parts that are mounted on the dash. The front side of the dash is always that next the engine.

Where parts are numbered, No. 1 should be toward the front of the car. For instance, No. 1 cylinder is the one nearest the radiator (in conventional construction).

"Right" and "left" are to the right- and left-hands when sitting facing forward in one of the seats of the car.

Studs, screws and bolts should take names from parts they serve to hold in place, although they are assembled with other parts. For example, the cylinder stud is permanently screwed into the crankcase but holds the cylinder in place.

The name "engine" should be used rather than "motor" to avoid confusion with electric motors.

DIVISION I—CYLINDERS

Group 1—Cylinders
Cylinder
L-head cylinder (valves on one side of cylinder)
T-head cylinder (valves on opposite sides of cylinder)
I-head cylinder (valves in cylinder head)
F-head cylinder (one valve in head; other on side, directly operated)
("Cast in block", not "cast en bloc")
(Cylinders of V-type engines should be numbered IR, IL, 2R, etc.)

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Group 1—Cylinders—Continued

Inlet-valve cap
Exhaust-valve cap
Cylinder-head gasket
Cylinder-head plug
Water-jacket top cover
Water-jacket top cover gasket
Water-jacket side (or front or rear) cover
Valve-spring cover
Valve-spring-cover gasket
Valve-spring-cover stud
Valve-stem guide
Priming-cup

Group 2—Crankcase

Crankcase
Barrel-type crankcase
Split-type crankcase (split horizontally, at or near center line of crankshaft)
Crankcase upper half
Crankcase lower half (used only when the lower half contains bearings. A crankcase of either barrel or split type, in which all the bearings are mounted directly on the part to which the cylinders are attached, is called a "crankcase," the terms "upper half" and "lower half" not being used)
Oil-pan (used for lower part of split-type or barrel-type crankcase, whether this serves as an oil reservoir or not)
Oil-pan drain-cock (or -plug)
Breather
Oil-pan gasket
"Bushing" instead of "bearing" for removable and renewable lining used in a plain bearing.
Crankshaft front bearing bushing (upper half and lower half)
Crankshaft front bearing cap
Crankshaft front bushing support (sometimes used in barrel-type crankcase)
Crankshaft rear bearing bushing
Crankshaft rear bearing shims (other shims accordingly)
Crankshaft center bearing bushing (if only three bearings or if all except end bearings are alike)
Crankshaft second bearing bushing, etc. (if more than three bearings; for example, front bearing, second bearing, third bearing, fourth bearing, rear bearing)
Hand-hole cover
Hand-hole-cover gasket
Timing-gear cover
Timing-gear-cover gasket
Flywheel housing
Generator bracket (other brackets take name of part supported)

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Group 3—Crankshaft
- Crankshaft
- Flywheel
- Crankshaft timing-gear (or -sprocket)
- Crankshaft timing-gear key
- Flywheel starter-gear
- Crankshaft starter-sprocket
- Flywheel studs
- Clutch-spring stud
- Crankshaft starting jaw (or pin)

Group 4—Starting-crank
- Starting-crank
- Starting-crank jaw
- Starting-crank shaft
- Starting-crank handle
- Starting-crank-handle pin

Group 5—Connecting-rods
- Connecting-rod
- Straight connecting-rod } V-type engine
- Forked connecting-rod } V-type engine
- Connecting-rod cap
- Connecting-rod bushing (upper half and lower half)
- Connecting-rod cap stud (or bolt)
- Connecting-rod cap nut
- Connecting-rod bearing shims
- Connecting-rod dipper
- Piston-pin bushing

Group 6—Pistons
- Piston
- Piston-pin
- Piston-pin lock-screw (in connecting-rod or piston)
- Piston-ring
- Piston-ring groove

DIVISION II—VALVES

Group 1—Camshaft
- Camshaft
- Eccentric shaft (Knight engine)
- Camshaft timing-gear
- Camshaft timing-gear key
- Camshaft idler gear
- Camshaft oil-pump gear
- Camshaft ignition-distributor gear
- Exhaust cam
- Inlet cam
- Oil-pump eccentric (or cam)

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Group 2—Valves

Valves should be numbered 1 Ex, 1 In, 2 Ex, 2 In, etc., according to the number of the cylinder. On V-type engines the numbers should be 1 REx, 1 LEx, etc.

Poppet valve
Inlet valve
Exhaust valve
Valve-spring
Valve-spring retainer
Valve-spring retainer lock
Valve-lifter
Valve-lifter guide
Valve-lifter-guide clamp
Valve-lifter roller
Valve-lifter-roller pin
Valve adjusting screw
Valve adjusting screw nut
Valve-rocker (either at cam or at overhead valve; if both, upper and lower)
Valve push-rod (intermediate between lifter and valve in I-head engine)

DIVISION III—COOLING SYSTEM

Group 1—Fan

Fan
Stationary fan support
Adjustable fan support
Fan hub
Fan-blades
Fan pulley
Fan-belt
Fan driving pulley

Group 2—Radiator

Radiator core
Radiator shell
Radiator upper tank
Radiator right side
Radiator left side
Radiator lower tank
Radiator filler-cap
Radiator strainer
Radiator drain-cock

Group 3—Pump

Water-pump
Water-pump impeller
Water-pump-impeller key
Water-pump body (in case of doubt, body is member mounted on engine)
Water-pump cover

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Water-pump shaft
Water-pump gland (part in contact with packing, whether threaded or not)
Water-pump-gland nut (or screw, or other part used to compress gland)
Water-pump shaft gear

Group 4—Pipes and Hose
Engine water outlet
Engine water inlet
Radiator hose (upper and lower)
Radiator water fitting (upper and lower)
Water-pump outlet pipe

DIVISION IV—FUEL SYSTEM

Group 1—Carbureter and Inlet Pipe
Carbureter
Inlet manifold (more than one connection to cylinder)
Inlet pipe (only one connection to cylinder)
Inlet manifold or pipe gaskets (at cylinders)
Carbureter gasket

Group 2—Carbureter Control
(Throttle control rods will take names from parts they connect, shafts by location or arrangement, and brackets by parts they support)
Accelerator pedal
Accelerator-pedal bracket
Accelerator-pedal pin
Accelerator-pedal rod
Accelerator-pedal rod-end pin
Carbureter mixture hand-regulator
Carbureter choke

Group 3—Carbureter Air-heater
Carbureter air-heater
Carbureter hot-air pipe

Group 4—Fuel Tank
Fuel tank
Fuel reserve tank
Fuel gage
Fuel-gage float
Fuel-gage glass
Fuel-tank outlet strainer
Fuel-tank outlet (flange, fitting, etc.)
Fuel-tank pressure flange (or fitting)

Group 5—Fuel Pipes and Feed Systems
Main fuel valve
Reserve fuel valve
Fuel pipe, main tank to auxiliary tank (or names of other parts connected)

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Group 5—Fuel Pipes and Feed Systems—Continued

Fuel pressure-pump (power pump)
Fuel hand-pump
Fuel pressure-gage pipe
Fuel pressure-gage tee
Fuel pressure pipe to tank
Fuel pressure-pump pipe
Fuel hand-pump pipe
Fuel hand-pump tee
Fuel pressure gage

DIVISION V—EXHAUST SYSTEM

Group 1—Exhaust Manifold
Exhaust manifold
Exhaust manifold gasket

Group 2—Exhaust Pipe and Muffler
Muffler
Exhaust pipe (extends from exhaust manifold to muffler. If in more than one part name sections front and rear. For V-type engines with two pipes, name right and left)
Muffler outlet pipe

DIVISION VI—LUBRICATION SYSTEM

Group 1—Oil-pan or Reservoir
Oil-pan
Oil tank (when separate)
Oil-filler strainer
Oil-filler cap

Group 2—Oil-pump
Oil-pump
Oil-pump body (any type of pump)
Oil-pump plunger
Oil-pump-plunger spring
Oil-pump inlet valve
Oil-pump outlet valve
Oil-pump shaft
Oil-pump shaft gear (outside the pump)
Oil-pumping shaft gear (inside the pump)
Oil-pumping follower gear
Oil-pump cover

Group 3—Oil Pipes, Strainers, Gages
(Oil pipes should be named from the parts they connect, as "Oil-pump to pressure-gage pipe")
Circulating-oil strainer
Oil strainer cap
Sight feed
Sight-feed glass
Oil level-gage
Oil level-gage float
Oil level-gage glass
Oil pressure-gage

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DIVISION VII—IGNITION

Group 1—Spark-plugs, Cables and Switches
- Spark-plugs
- Spark-plug cables (numbered according to cylinders)
- Coil high-tension cable
- (Low-tension cables should be named from the parts they connect, as: "Storage battery to ignition switch cable." In case of more than one conductor the cable should be designated as double, triple, etc.)
- Ignition coil
- Ignition switch
- Dry cell (two or more cells make a dry battery)

Group 2—Ignition Distributor
- Ignition-distributor breaker
- Ignition-distributor breaker-arm
- Ignition-distributor breaker-arm point
- Ignition-distributor fixed breaker-point
- Ignition-distributor brush
- Ignition-distributor shaft
- Ignition-distributor shaft gear

Group 3—Magneto
- Magneto
- Magneto distributor
- Magneto breaker-box
- Magneto breaker-arm
- Magneto fixed breaker-point
- Magneto breaker-arm point
- Magneto distributor brush
- Magneto collector-ring brush
- Magneto coupling, pump end
- Magneto coupling, center member
- Magneto coupling, magneto end

Group 4—Ignition Control
- Spark control rod (name parts connected)
- (Other control parts named as explained under throttle control)

DIVISION VIII—STARTING AND LIGHTING EQUIPMENT

General

A one-unit system uses a starter-generator.

A two-unit system uses a generator and a starting motor.

A combined unit system uses a duplex starter-generator.

Group 1—Generator
- Generator
- Generator brush
- Generator brush-holder
- Generator gear
- Generator shaft
- Generator coupling (members as indicated under magneto coupling)

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Group 2—Starting Motor
- Starting motor
- Starting-motor brush
- Starting-motor brush-holder
- Starting-motor pinion
- Starting-motor intermediate gear
- Starting-motor intermediate-gear shaft
- Starting-motor intermediate pinion
- Overrunning clutch

Group 3—Wiring
- (Cables and conduits should be named from parts they connect)
- Starting switch
- Starting-switch pedal (or lever)

Group 4—Battery
- Storage battery
- Filler cap
- Terminal post
- Connector strip

DIVISION IX—MISCELLANEOUS ELECTRICAL EQUIPMENT

Group 1—Lamps and Wiring
- Head-lamp
- Tail-lamp
- Side-lamp
- Instrument lamp
- Tonneau lamp
- Dome lamp
- Pillar lamp
- Inspection lamp
- Inspection-lamp cord
- Inspection-lamp plug
- Inspection-lamp socket
- Head-lamp socket
- Head-lamp support
- Head-lamp support tie-rod
- Tail-lamp support
- (Cables and conduits should be named from the parts they connect)
- Junction-box (wires not attached to box)
- Junction-box screw
- Junction-box cover
- Fuse-box
- Fuse-box cover
- Fuse-block
- Fuse-clip
- Fuse (designated by name of part fed by circuit)
- Junction panel

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Group 2—Switches and Instruments
- Lighting switch
- Ammeter
- Voltmeter
- Volt-ammeter
- Charging indicator
- Reverse-current cutout
- Current regulator

Group 3—Horn
- (No names have been selected for horn parts)

Group 4—Miscellaneous
- (Will include any additional electrical equipment such as electrical gearshift)

DIVISION X—CLUTCH

General

- Plate clutch (one plate clamped between two others)
- Disk clutch (more than three disks)
- Dry disk clutch
- Lubricated disk clutch
- Cone clutch (leather faced, asbestos faced)
- Expanding clutch

Group 1—Clutching Parts

Cone Clutch

- Clutch cone
- Clutch facing
- Clutch-facing spring
- Clutch-facing-spring plunger
- Clutch spring
- Clutch thrust-bearing
- Clutch cone hub
- Clutch cone bushing
- Clutch-spring spider (for cone clutch with multiple springs)
- Clutch-spring stud
- Clutch-spring retainer
- Clutch-spring nut
- Clutch spindle
- Clutch shaft (not attached to crankshaft)
- Clutch shaft bearing (not in transmission case)

Disk Clutch

- Clutch case (rotating member)
- Clutch housing (non-rotating member)
- Clutch cover
- Clutch housing cover
- Clutch driving disk
- Clutch driven disk

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Group 1—Clutching Parts—Continued

Clutch driving disk stud
Clutch pressure plate (front and rear, if two—used on both disk and plate clutches)
Clutch driven spider (or drum—driving and driven if two)
Clutch cork-inserts
(Facing, spring, thrust-bearing, etc., as under cone clutch)

Plate Clutch

Clutch driven plate
Clutch driving plate
Clutch pressure levers
(Other parts as under cone and disk clutches)

Group 2—Releasing Parts

Clutch release sleeve
Clutch release shoe or clutch release bearing housing
Clutch release bearing
Clutch release fork
Clutch release fork shaft
Clutch pedal shaft
Clutch pedal adjusting link
Clutch release fork lever
Clutch pedal
Clutch pedal pad
Clutch brake
Clutch brake facing

DIVISION XI—TRANSMISSION

Group 1—Transmission

Transmission case (upper half and lower half, if bearings seat in both)
Transmission case cover
Clutch gear
Clutch gear bearing (front and rear if two)
Clutch gear bearing retainer
Countershaft
Countershaft front bearing (if ball or roller)
Countershaft front bearing bushing (if plain bearing)
Countershaft front bearing retainer
Countershaft rear bearing retainer
Countershaft drive gear
Countershaft second-speed gear
Countershaft low-speed gear
Countershaft reverse gear
Reverse idler gear
Reverse idler gear shaft
Reverse idler gear bushing

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Transmission shaft
Transmission shaft pilot bearing
Transmission shaft pilot bearing bushing (if plain)
Transmission shaft rear bearing
Transmission shaft rear bearing retainer
Second and high sliding gear
Low and reverse sliding gear

Group 2—Shifting Mechanism
High-gear shift fork
Low-gear shift fork
Reverse shift fork (if three are used)
High-gear shift bar
Low-gear shift bar
Reverse shift bar

Group 3—Control
Gearshift bar selector
Gearshift lever shaft
Low-gear shift connecting-rod
High-gear shift connecting-rod
Gearshift hand lever ("hand" may be omitted)
Gearshift hand lever bracket ("hand" may be omitted)
Gearshift housing (center control)
Gearshift gate

Group 4—Propeller-shaft
Propeller-shaft
Propeller-shaft front universal-joint (assembly—"propeller-shaft" may be omitted)
Propeller-shaft rear universal-joint (assembly—"propeller-shaft" may be omitted)
Propeller-shaft front bearing (with enclosed shaft)
Transmission shaft universal-joint flange (substitute name of any other shaft on which flange is mounted)
Universal-joint flange yoke
Universal-joint slip yoke
Universal-joint plain yoke
Universal-joint center cross (ring or block)
Universal-joint bearing bushing
Universal-joint pin (may be designated as long and short, straight and shoulder, etc.)
Universal-joint inner casing
Universal-joint outer casing
Universal-joint casing packing
Universal-joint casing nut
Universal-joint trunnion (for trunnion type joint)
Universal-joint trunnion block

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DIVISION XII—REAR AXLE

General Types

Dead Axle—An axle carrying road wheels with no provision in the axle itself for driving them.

Live Axle—General name for type of axle with concentric driving shaft.

Plain Live Axle—Has shafts supported directly in bearings at center and at ends, carrying differential and road wheels.

(The plain live axle is practically extinct.)

Semi-Floating Axle—Has differential carried on separate bearings, the inner ends of the shafts being carried by the differential side gears, and the outer ends supported in bearings.

The semi-floating axle shaft carries torsion, bending moment, and shear. It also carries tension and compression if the wheel bearings do not take thrust, and compression if they take thrust in only one direction.

Three-Quarter Floating Axle—Inner ends of shafts carried as in semi-floating axle. Outer ends supported by wheels, which depend on shafts for alignment. Only one bearing is used in each wheel hub.

The three-quarter floating axle shaft carries torsion and the bending moment imposed by the wheel on corners and uneven road surfaces. It also carries tension and compression if the wheel bearings are not arranged to take thrust.

Full-Floating Axle—Same as three-quarter floating axle except that each wheel has two bearings and does not depend on shaft for alignment. The wheel may be driven by a flange or a jaw clutch.

The full-floating axle shaft is relieved from all strains except torsion, and in one possible construction, tension and compression.

Types of Axle Drive

The different types of live axle can be driven by *Bevel Gear, Spiral Bevel Gear, Worm, Double-reduction Gear* or *Single Chain.*

In other constructions, the rear wheels are driven by *Double Chains, Internal Gears,* or *Jointed Cross-shaft.*

Group 1—Housing

Rear-axle housing (if one piece)
Right and left halves (if two pieces)
Bevel (or worm) gear housing }
Right rear-axle tube } (if three pieces)
Left rear-axle tube }
Rear-axle-housing cover
Differential carrier (bolted to housing)
Rear-axle spring-seat
Axle brake-shaft bracket (right and left)

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Wheel brake-support, right and left ("wheel" may be omitted)
Wheel brake-shield ("wheel" may be omitted)

Group 2—Torque-arm and Radius-rod
Radius-rods

Group 3—Drive Pinion
Axle drive bevel pinion (or worm)
Axle drive pinion (or worm) shaft
Axle drive pinion front bearing
Axle drive pinion rear bearing
Axle drive pinion thrust-bearing
Axle drive pinion front bearing adjuster
Axle drive pinion front bearing adjuster lock
Axle drive pinion rear bearing adjuster
Axle drive pinion rear bearing adjuster lock
Axle drive pinion adjusting sleeve (containing both bearings)
Axle drive pinion (or worm) carrier

Group 4—Differential
Axle drive bevel (or worm) gear
Differential
Differential case, right
Differential case, left
Differential side gear
Differential spider pinion ("spider" may be omitted)
Differential spider (or pinion shaft)
Differential bearing
Differential thrust-bearing
Differential bearing adjuster
Differential bearing adjuster lock

Group 5—Axle Shafts
Axle shaft (right and left)
Axle shaft wheel-flange (or clutch)

DIVISION XIII—BRAKES

General

In the following list of brake parts the terms "outer" and "inner" are used, being applicable to any case of two sets of brakes on the rear wheels. Where the brakes are external and internal these terms may be substituted for "outer" and "inner." Where one brake is located at the wheels and the other at the transmission the terms "wheel brake" and "transmission brake" should be substituted. With other concentric or side-by-side brakes the terms "outer" and "inner" should be retained, "outer" indicating in the latter case the ones nearer the wheels.

The list is made up for external contracting and internal expanding brakes. If both brakes are of one type the necessary changes will be obvious. The designation of brake parts on the rear axle

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as foot-brake or hand-brake parts, or by equivalent terms, is too remote to be clear, especially in the case of stock axles whose brakes may be connected either way according to chassis design. Nearly the same condition prevails in regard to designating parts on the chassis according to whether they are connected to the inner or outer brakes at the axle.

The terms "service brake" and "emergency brake" should not be used. Better designations are "foot brake" and "hand brake"; or if both are foot-operated, "right foot-brake" and "left foot-brake."

Group 1—Outer Brake
- Outer brake band
- Outer brake band lining
- Outer brake band adjusting nut (yoke, etc.)
- Outer brake hand lever
- Outer brake lever shaft
- Outer brake shaft inner end lever
- Outer brake shaft outer end lever

Group 2—Inner Brake
- Inner brake shoe (or band)
- Inner brake shoe (or band) lining
- Inner brake toggle (link, etc.)
- Inner brake toggle lever
- Inner brake toggle shaft
- Inner brake cam
- Inner brake camshaft
- Inner brake camshaft (or toggle shaft) lever

Group 3—Pedal (or outer) Brake Control
- Outer brake rod
- Outer brake rod yoke
- Outer brake intermediate shaft (or tube)—right and left
- Outer brake intermediate shaft (or tube)—right lever
- Outer brake intermediate shaft (or tube)—left lever
- Outer brake intermediate shaft (or tube)—center lever
- Outer brake right equalizer lever
- Outer brake left equalizer lever
- Outer brake equalizer
- Brake pedal
- Brake pedal rod
- Brake pedal rod yoke
- Brake pedal pad
- Brake pedal shaft

Group 4—Hand (or inner) Brake Control
- Inner brake rod
- Inner brake rod yoke
- Inner brake intermediate shaft (or tube)—right and left
- Inner brake intermediate shaft (or tube)—right lever
- Inner brake intermediate shaft (or tube)—left lever

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Inner brake intermediate shaft (or tube)—center lever
Inner brake right equalizer lever
Inner brake left equalizer lever
Inner brake equalizer
Brake hand lever rod
Brake hand lever rod yoke
Brake hand lever
Brake lever segment (or sector)
Brake lever pawl
Brake pawl spring
Brake pawl button
Brake pawl finger lever
Brake pawl rod

DIVISION XIV—FRONT AXLE AND STEERING

Group 1—Axle Center
Front axle center
Front spring seats
Front axle bushing

Group 2—Steering-knuckles
Right steering-knuckle
Left steering-knuckle
Steering-knuckle bushing (upper and lower)
Steering-knuckle pivot
Steering-knuckle-pivot nut
Steering-knuckle thrust-bearing
Right steering-knuckle arm
Left steering-knuckle arm
Steering-knuckle gear rod arm

Group 3—Steering-rods
Steering-knuckle tie-rod
Steering-knuckle tie-rod end
Steering-knuckle tie-rod clamp bolt
Steering-knuckle tie-rod pin
Steering-gear connecting-rod

Group 4—Steering-gear
Steering-gear case
Steering-gear-case cover
Steering-gear bracket
Steering-gear arm
Steering-arm shaft (if separate from sector or other operating member)
Steering-wheel rim
Steering-wheel spider
Steering-wheel tube (or shaft)
Spark and throttle sector
Spark and throttle sector tube

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Group 4—Steering-gear—Continued.

Spark hand-lever
Spark hand-lever tube (or rod)
Throttle hand-lever
Throttle hand-lever tube (or rod)
Steering-column tube (stationary)
Steering-column cowl (or dash or floor) bracket

The various bushings in the steering-column take names from parts to which they are permanently fitted, being further distinguished as upper and lower, inner and outer, if necessary. Bushings in the steering-gear case take names from the worm and sector or other main operating parts which they support, as: Steering-gear worm upper bushing; although the steering-wheel tube may be the member which turns inside the bushing.

Steering worm
Steering-worm sector (or gear) } (worm and sector gear)
Steering-worm shaft

DIVISION XV—WHEELS

Group 1—Front Wheels

Front wheel felloe
Front wheel felloe band
Front wheel rim
Rim bolts
Rim clamps
Front wheel hub
Front wheel hub-flanges
Front wheel hub-cap
Front wheel outer bearing
Front wheel outer bearing inner race
Front wheel outer bearing outer race
Front wheel outer bearing balls
Front wheel outer bearing ball retainer
Front wheel outer bearing rollers
Front wheel outer bearing roller cage
Front wheel inner bearing (parts same as outer bearing)
Front wheel bearing spacer
Front wheel bearing nut
Front wheel bearing lock nut
Front wheel bearing locking washer

Group 2—Rear Wheels

Rear wheel hub
Rear wheel hub-flange
Rear wheel hub-cap
Rear wheel outer bearing
Rear wheel inner bearing
Wheel brake-drum
(Other parts named like front wheel parts)

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DIVISION XVI—FRAME AND SPRINGS

Group 1—Frame
- Frame side member (right and left)
- Front cross member
- Rear cross member
- Center cross member
- (As above if only three cross members, as below if more than three)
- First cross member
- Second cross member, etc.
- Sub-frame side member (right and left)
- Sub-frame cross member (front and rear)
- Right rear gusset (upper and lower)
- (Gussets at other cross members named according to member)

Group 2—Frame Brackets and Sockets
- Front spring front bracket (right and left)
- Front spring rear bracket (right and left)
- Rear spring front bracket (right and left)
- Rear spring rear bracket (right and left)
- Running-board bracket (front, right, etc., if not duplicates)
- Running-board bracket brace
- Engine front support bracket
- Engine rear support bracket
- Torque-arm bracket
- Radius-rod bracket

Group 3—Front Springs
- Front spring (right and left)
- Front spring shackle
- Front spring shackle-bolt (upper and lower)
- Front spring front bolt
- Front spring rebound-clip
- Front spring seat
- Front spring seat pad
- Front spring clip
- Front spring clip plate
- Front spring center-bolt

Group 4—Rear Springs
- Rear springs (upper and lower for elliptic and three-quarter elliptic)
- Rear spring pivot bolt (or pin) } (for half-elliptic cantilever spring)
- Rear spring pivot seat } (for half-elliptic cantilever spring)
- Rear spring double shackle } (for platform spring)
- Rear side spring } (for platform spring)
- Cross spring } (for platform spring)
- (Other parts as for front springs)

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DIVISION XVII—HOOD, FENDERS AND SHIELDS

Group 1—Hood
- Hood
- Hood sill
- Hood handle
- Hood fastener
- Hood fastener bracket (spring, lever, etc.)

Group 2—Engine Shield
- Engine shield
- Engine shield fastener
- Engine shield bracket (spring, etc.)

Group 3—Fenders and Running-boards
- Running-board (right and left)
- Running-board linoleum covering
- Running-board outside binding
- Running-board inside binding
- Running-board front binding
- Running-board rear binding
- Running-board shield (right and left)
- Right front fender
- Left front fender
- Right rear fender
- Left rear fender
- Fender support socket
- Right front fender front support
- Right front fender rear support
- (Other fender supports accordingly)

Group 4—Windshield*

DIVISION XVIII—BODY

Types of Bodies

Roadster—An open car seating two or three. It may have additional seats on running-boards or in rear deck.

Coupelet—Seats two or three. It has a folding top and full-height doors with disappearing panels of glass.

Coupe—An inside operated, enclosed car seating two or three. A fourth seat facing backward is sometimes added.

Convertible Coupe—A roadster provided with a detachable coupe top.

Clover Leaf—An open car seating three or four. The rear seat is close to the divided front seat and entrance is only through doors in front of the front seat.

*Names for windshield parts have not been selected.

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Touring Car—An open car seating four or more with direct entrance to tonneau.

Salon Touring Car—A touring car with passage between front seats, with or without separate entrance to front seats.

Convertible Touring Car—A touring car with folding top and disappearing or removable glass sides.

Sedan—A closed car seating four or more all in one compartment.

Convertible Sedan—A salon touring car provided with a detachable sedan top.

Open Sedan—A sedan so constructed that the sides can be removed or stowed so as to leave the space entirely clear from the glass front to the back.

Limousine—A closed car seating three to five inside, with driver's seat outside, covered with a roof.

Open Limousine—A touring car with permanent standing top and disappearing or removable glass sides.

Berline—A limousine having the driver's seat entirely inclosed.

Brougham—A limousine with no roof over the driver's seat.

Landaulet—A closed car with folding top, seats for three or more inside, and driver's seat outside.

Group 1—Floor-boards and Dash
- Floor-boards (horizontal)
- Toe-boards (sloping)
- Heel-boards (under seats)
- Dash (separates engine compartment from driver's compartment)
- Instrument board

Group 2—Body*

Group 3—Upholstering*

Group 4—Top*

DIVISION XIX—ACCESSORIES

Group 1—Speedometer*

Group 2—Tire-pump
- Tire-pump
- Tire-pump driving gear
- Tire-pump shaft gear
- Tire-pump idler gear

Respectfully submitted.

NOMENCLATURE DIVISION,

K. W. ZIMMERSCHIED,

Chairman.

*Names for parts in these groups have not been selected.

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DISCUSSION

K. W. ZIMMERSCHIED:—The work of giving the principal parts of automobiles standard names was started in 1911, but has progressed little since its inception. One of the great difficulties in the earlier days arose from the fact that there were so many types of automobiles, engines, transmissions and suspensions, in which the details varied so greatly from make to make that it seemed practically impossible to get any great number of terms that would apply to a large number of different makes of cars.

As a result of evolution, however, designs have settled down more and more into a few general types, so that we have been enabled in the last year or so to get up this list of standard names as a skeleton or framework that any automobile manufacturer can use as a foundation for his parts list. We also trust that engineers will familiarize themselves with this list so that they will use the terms widely and thus make our nomenclature practice as uniform as possible.

HENRY SOUTHER:—The thanks of the Committee are due to the gentlemen who completed this work. I had something to do with it a good many years ago, and I know what it means. Any one standard list, even if it is not perfect, is better than many of them.

(On motion, duly seconded, it was voted to approve the report of the Nomenclature Division for submission to the Society for adoption.)

FORMATION OF AERONAUTIC ENGINE DIVISION

CHAIRMAN CLAYDEN:—We would like to hear from Mr. Souther regarding the new Division we are about to establish, the Aeronautic Engine Division.

HENRY SOUTHER:—The need for such a Division has arisen through the possibility of direct connection between the Army and Navy and the manufacturing and engineering activities represented by this Society. Because of the formation of the National Advisory Committee for Aeronautics, which is authorized by Congress, and reports directly to the President, it is now possible for the Government and this engineering body to work together. The Government will be represented by an officer of the Army, an officer of the Navy and an official of the Bureau of Standards; consequently the whole force of Government organization can be put behind the movement.

The intention is for the United Service representatives to fight their own battles, prepare drawings, and give them to the Division for criticism from a manufacturing and from an engineering standpoint. As soon as an agreement is reached between the two committees the United Service will standardize. That procedure will mean rapid

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progress because the authorities in Washington demand that there shall be standards and that they shall be used.

We need standard gasoline pipes, so that when one plane is wrecked its parts can be used in another plane that may be wrecked also; we can then put the parts of the two planes together and get a new one. At the front in Europe if a plane is completely wrecked, the engine with its auxiliary parts is taken out, put into a standard box and shipped at once to a base repair station. There it is immediately taken into a storehouse, pulled to pieces, inventoried, repaired and sent to the front and used again.

CHAIRMAN CLAYDEN:—This new Division of the Standards Committee is to be under the chairmanship of Mr. Souther, and will call for assistance from other Divisions. The Engine and Transmission Division will have certain demands made upon it. The Carbureter Fittings Division and possibly the Chain and other Divisions will have to lend a hand.

DISCUSSION OF STANDARDS COMMITTEE REPORTS AT SOCIETY MEETING

The Division reports approved at the Standards Committee meeting June 12 were submitted to the Society at a session held on June 15. Reports from the Ball and Roller Bearings, Carbureter Fittings, Electrical Equipment, Electric Vehicle, Iron and Steel, Miscellaneous, and Nomenclature Divisions were then accepted for submission to the members by letter ballot. These reports were accepted without extended comment, with the exception of those submitted by the Electrical Equipment and Nomenclature Divisions.

REPORT OF ELECTRICAL EQUIPMENT DIVISION*

A. D. T. LIBBY:—Before this report is voted on, it might be well if some of the details of the lamp bracket were explained—why it was thought that this particular type is to be recommended for standard practice.

CHAIRMAN CLAYDEN:—The Division believed that the single standard type is in process of development, and that at the present time we are not prepared to handle that. A great number of fork-type brackets are in use, differing in size by small amounts. It was obvious that three standardized forks would cover all sizes of fork support except those for large lamps for big and costly cars, or for lamps having an outside shell of distinctive pattern. The acceptance of these three fork-type lamp supports should not be taken as in any way discouraging the adoption of the single type. They are merely recommended for use where it is desired to use a fork pattern.

*See page 6 for final form of report.

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A. D. T. LIBBY:—The whole idea of the Division has been to recommend a support in order to standardize the location of the bulb in the lamp; the standardization of the filament, the bulb and its filament is already under consideration. It was felt that with this type of support there would not be as much variation in the location and adjustment of the beam of light as there would be were the lamp fastened with a set-screw on the side. That is the reason for having the threaded end, with nut and lock washer to hold the lamp in place.

The Division is working to standardize the position of the filament in the bulb, and then the position of the bulb itself in the lamp. It is believed that when these matters are settled the support dimensions will meet the requirements. As the forks are made adjustable at the base, they will not have to be bent for adjustment, and the whole non-glare situation will be taken care of.

REPORT OF NOMENCLATURE DIVISION*

K. W. ZIMMERSCHIED:—The report contains an introductory discussion of the way in which we have gone about the work of getting more uniformity in practice. This discussion should be read thoroughly before you criticize the names in the list itself. There are doubtless many decisions with which you will not agree, but if you will bear in mind the introduction and remember that no name has been selected without most careful scrutiny you will, I think, find justification for most of the decisions. Wherever there was a choice between several words, we tried to use that which was as fairly descriptive as possible of the function of the piece, bearing in mind present usage, accuracy and general distinctiveness.

The names are divided among several general divisions, which are grouped in such a way that several of them can be combined to form one portion of a catalog or parts list. For instance, the divisions from I to and including IX have to do with the production of power. Those from X to and including XV include practically all the parts that have to do with the transmission of the power produced by the first nine groups; and finally, divisions XVI to XIX inclusive take care of all those parts that have to do with the accommodation of passengers. This classification appealed to nearly all of the Division members as the most logical foundation on which to build up a parts list.

From some viewpoints the system of grouping together those parts that are assembled together has advantages, but, everything considered, from the viewpoint of the problems before the Nomenclature Division and from the viewpoint of final satisfaction to the customer or the factory, grouping by assemblies seems to be less advantageous than grouping by functions.

Probably the greatest difficulty came in deciding upon the names for body types. In the absence of any clear and authoritative defini-

*See page 28 for final form of report.

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tions, we have determined upon those in this list and trust that they will have some influence in bringing order out of the present chaos.

We hope the Society will be able to publish this list at some future time in a much more elaborate form, including illustrations of typical examples of the parts named.

A. J. SLADE:—I would like to inquire whether the Nomenclature Division, after disposing of the nomenclature for passenger vehicles, which this report appears to do, intends to add nomenclature for such parts as are distinctive in merchandise-carrying vehicles and also those distinctive in electric vehicles. The electric vehicle, especially, has many parts that do not appear in this list. I presume that eventually the list of names will cover all types of vehicles in which the membership of the Society is interested.

K. W. ZIMMERSCHIED:—That depends very largely on the reception of what we have done. If we receive sufficient encouragement we will be glad to go ahead with the work to any extent that is necessary.

Names of Ignition Parts

A. D. T. LIBBY:—In Group 2, Division VII, "timer-distributor" is preferable to "ignition-distributor," particularly as in Group 3 "magneto" also refers to ignition.

K. W. ZIMMERSCHIED:—We fought for two hours to decide whether that name should be "timer-distributor" or "ignition-distributor." There is so much variation in the principles on which these distributors are built that we went back to what the thing does; that is, distributes the ignition current. The timing is merely incidental to the proper distribution.

CHAIRMAN CLAYDEN:—This discussion shows anyone who has not attended a meeting of the Nomenclature Division the type of argument that took place. The case of the ignition-distributor is a particularly good one, because it emphasizes the fact that wherever there was considerable disagreement, as there was over this name, unanimity was obtained by taking the most fundamental word we could find, the most descriptive word, or the word that was the most accurate—the nearest to scientific correctness. That system has been followed all through the list and the word that seemed to have the widest and most accurate meaning has been chosen, except in cases where it was completely different from general practice.

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BUSINESS AND PROFESSIONAL SESSIONS OF SOCIETY MEETING

PRESIDENTIAL ADDRESS BY RUSSELL HUFF

The Society is enjoying a most favorable condition of affairs. In spite of unexpected expenses developing since the annual budget was prepared, the income due to increased membership has more than kept pace with the increased expenses, and to-day we show a favorable balance over and above the expected requirements. We want to maintain this balance in the form of a reserve fund to be set aside to meet unexpected future contingencies. In this connection, I want to mention the able assistance rendered the Society by the Finance Committee, whose valuable advice has been instrumental in having the Society's funds safely handled and invested.

Much praise is due the Membership Committee, which has taken hold of its work in an energetic and business-like way and which has increased our membership approximately 150 since January. There are now 1910 members, with 66 applications awaiting action of the Council. While this is a reassuring sign of the high esteem engineers have for our Society, it does not mean that all the country's eligible men have been nominated and elected to membership.

There are now about 900 companies represented by our members, but there are approximately 5000 firms engaged in the manufacture of motor cars, trucks, tractors, aeroplanes, marine motors and various kinds of accessories.

We have some companies who have ten men in our Society, but our actual average is 2.2 men per company. Taking one as a fair average for the number of men from each company who should belong to the Society, our membership should be about 5000 instead of 2000. I am anxious to interest you in these figures and to enlist your enthusiastic support in assisting the Membership Committee to secure these additional prospects.

I feel that the Society could accomplish a greater good in the pursuit of its many lines of endeavor if it had all the companies allied to the industry represented on its membership list.

It is not necessary to review in detail the many points of mutual interest and the close relationship of the engineers in the automobile, aero, marine, tractor and stationary gas engine field, except, suffice it to say, that important problems such as fuel, engine lubrication, ignition, thermal efficiency, mechanical efficiency, balance, cooling,

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cost, weight and many other such phases of engineering study are common to all the foregoing lines of development.

Why then should not the Society which is fostering the study of all these items be the parent national body to which all the engineers in their allied industries should belong?

A strong and active Meetings Committee has been appointed for the year, and we will have a chance to measure its ability during this trip, because all the details of this cruise and convention have been arranged by this committee. It is difficult for anyone not connected with the work to realize the immense amount of detail involved in planning a trip of this kind and successfully carrying out its details. The auspicious start which has characterized the trip up to this moment augurs well for its success.

The Papers Committee deserves special mention on account of the excellent manner in which it, cooperating with the Meetings Committee, has edited and published the several papers to be presented at the sessions on this trip.

Unusual interest in the Society's work has prevailed throughout the six Sections during the year. Attendance at Sections meetings has steadily increased. This is due, no doubt, to the improvement in the papers presented. Your president feels that the work of your Sections is vital to the success of the main Society.

Since our Winter Meeting, the War Department staff at Washington has called upon us twice for conferences on military transportation problems and truck standardization work. A special committee, consisting of our general manager, C. F. Clarkson, as chairman, and Henry Souther, G. W. Dunham and W. P. Kennedy, has been appointed to cooperate with the Government on any military transportation problems that might arise. Another special temporary committee, composed chiefly of the engineers from the leading truck companies, had several meetings with representatives from the Quartermaster General's Department, completely reviewed the 1½-ton truck specifications and put them in a much more practical shape than would have been possible otherwise.

The Council has passed a resolution establishing a new division to study aviation problems.

We have recently been approached by the representatives of some of our allied engineering societies with a view of working out a satisfactory plan whereby their societies could be merged into the Society of Automobile Engineers. We believe that satisfactory arrangements will soon be made, whereby our Society will receive a greatly increased membership from engineers in the tractor, marine and areonautic field.

It is gratifying to see the broad influence our Society is wielding throughout the country and to see the Government's confidence in our body.

We can all well be proud of our Standards Committee. One hundred and twenty men, divided into fourteen divisions, serve in carrying

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on this valuable work. The industry is developing so rapidly that constant investigation and research work is required to keep pace with the art. Even with the brightest minds among the engineers concentrated on this labor, the problem of standardization is never finished. New conditions are constantly arising to affect the ideals of yesterday. The race among manufacturers for the most popular and easiest selling design has spurred engineers to develop new constructions undreamed of a few years ago. The annual flood of new models incorporating new and diversified engineering is a fertile field for additional standardization efforts.

While your president realizes that the Society has been doing invaluable work through its many lines of activities, he also feels that it is possible to accomplish still greater things and that there is a large field still untouched awaiting us.

THE AUTOMOBILE ENGINEER AND PREPAREDNESS

By Howard E. Coffin

(Past-president of the Society)

I have been asked to address this meeting on Industrial Preparedness. I do not altogether like the term preparedness. There is really a much better term, a term expressing that which must underlie any preparedness in this country. It is the term "prearrangement"; it is only through prearrangement and the most careful prearrangement in time of peace that we can hope to have any degree of true preparedness in time of war.

We are all agreed, I believe, that we seek an adequate defense in case of future aggression, and insofar as may be possible an insurance against any such aggression. It is a matter of common knowledge that we are not, and cannot become, a militaristic nation. Saber-clanking is not in our line, and never will be, for any great length of time. It should be clear to us, therefore, that if we are to adopt any policy of preparedness and expect it to be maintained through years of peace, as it must be if we are ever to be fitted to defend this country when war threatens, that policy must be one in keeping with the traditions and ideals of the nation.

It is easy now to appropriate hundreds of millions for armies and for battleships. It will not be so easy after peace has come again to the world. Yet Germany's wonderful state of preparedness was achieved during her long years of peace. The same great industrial

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machine with which she bade fair to conquer the commercial world is now making her seemingly unconquerable as a military power.

NEED FOR INDUSTRIAL ORGANIZATION

Every country of Europe has found it comparatively easy to supply and train millions of men for its armies. But Germany alone of all these nations was possessed of the organized industrial machinery for supplying the arms and munitions of war to her millions in the field. The present conflict has taught us a new conception of warfare. This war and all future wars will be decided by the fighting industries of the nations involved. Victory will go to the armies of the country that can most quickly and for the longest time supply its fighting line with the munitions of war. Victory will be won in the mills, the factories and the mines, far from the sound of the guns.

Industrial preparedness is basic preparedness. Two years of observation of the European war has taught us that organized industry is the foundation upon which we must rest any and every plan for military defense of this nation. Between 80 and 90 per cent of the manufacturing and producing resources of the European countries are engaged in the making of supplies for their armies. In the event that we are so unfortunate as to become involved in an argument with any first-class foreign power, an equal percentage, at least, of our manufacturing equipment will be needed in such service.

Industrial preparedness is strictly in keeping with the natural tendencies and abilities of our people. It is by far the cheapest form of preparedness. We have already the investments in plants, in tools, in machinery, and, more important still, our investment in skilled workers. A soldier can be made in a year—and millions of them have been—but it takes many years to make a toolmaker.

As I have said, we have all this equipment in our hands, but it is unorganized and uneducated for the national service. We cannot expect any big manufacturing plant to change over, even in time of peace, in less than one year's time, from its usual commercial line to a new product with which its departments and its workmen have had no previous experience. And yet that is exactly what at least 80 per cent of our plants will be called upon to do, under the most chaotic conditions, in case of a real war.

EFFECT OF WAR ON INDUSTRIAL MACHINE

We have heard of the difficulties that American manufacturers have had with foreign munitions orders. Many plants in this country are six months or a year behind their contract times in deliveries. This delay in delivery upon the part of these American companies has occurred at a time when the country has hardly been disturbed, at a time when raw material sources have been accessible, when transportation has not been interrupted. Compare the conditions under which our failures during the past two years have occurred

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with those that would have existed had we been one of the nations involved in the present war, with transportation lines tied up, sources of supply cut off, labor disorganized and much of it enlisting for service at the front.

I leave it to you to imagine the economic shock to our industrial machine immediately following a declaration of war under our present conditions. Let us take the experience in foreign countries eighteen months ago, and apply some of its lessons to our own situation. On Aug. 1, 1914, the wheels of peaceful commerce in Europe came to a stop. Few have since been started on peace products. In England and France the plants were closed down and manufacturing organizations built up during a long period of years were broken up. In both countries a large percentage of the skilled labor went into the armies and much of it to the front.

But it was not long before the awakening came. The wheels of industry were again started, but this time on the munitions of war. Motor-car factories were converted into shell factories, perhaps. Practically every line of industry found its equipment suited to the making of some article or part for the armies and navies. Watchmakers are making and adjusting shell fuses. Jewelry houses are making periscopes both for submarines and for making observations from the trenches. Manufacturers of baking machinery are turning out thousands of shells. One infants' food manufacturer is delivering millions of plugs for shells. Makers of textile machinery are building field kitchens. And so on through the thirty-odd thousand items needed by the fighting line. Whether it be toothbrushes or jackknives, rifles or cartridges, big guns or shells, foreign governments have learned that it is upon private industry that all hope of quantity production must be based. Government plants must become mere training schools and assembling depots in time of such national emergency. Four thousand private plants in England alone are being operated directly under government control. Many thousands of other private plants, both in England and in neutral countries, are working day and night on munitions of war for the English armies.

LABOR SITUATION IN COUNTRIES AT WAR

Meanwhile the labor situation in the countries involved is an interesting study. In France we have a military policy founded on universal service. A careful record is kept of every man, every French citizen, and of his experience and qualifications. In France, therefore, it was comparatively easy to bring men back from the front and set them again in their accustomed places at the lathe, the drill or the forge, from which they ought never to have been taken. Even in France, however, chaotic conditions resulted during a considerable and very dangerous period because of the disorganization of the industrial forces.

The situation in England should be of particular interest to us for two reasons: first, her traditions, conditions and governmental

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methods more nearly parallel our own; second, her policies, again probably parallel to our own under similar conditions, afford us a glaring example of what best not to do. No record of skilled labor was available, and thousands upon thousands of skilled workers have been wasted at the front who would have been worth their weight in gold in the factories at home.

In the meantime, some interesting things were happening in Germany. I cannot better illustrate than by quoting a story told me by an eye-witness. A telegram announcing the declaration of war was received by one large German manufacturing plant at 2 o'clock in the afternoon. Bells rang throughout the plant and the men filed past their pay windows. Slips of paper were given them carrying their instructions. One part of them left the plant at once on the way to concentration points; others proceeded to the storerooms in which were kept the gages, jigs and tools for use in the production of that material of war for which that great plant, through careful prearrangement, was to be held responsible. Others of the workmen left at once to report as experts in matters of ignition for aeroplanes and motor vehicles. The machinery scarcely even stopped. A few hours, at most, in shifting jigs and changing set-ups and the transition from the commercial product to the war product had been made. About one million dollars worth of jigs, gages, materials and tools pertaining to war product and owned by the German Government were stored in this one large plant alone. Here, in this simple story we find the meat of true preparedness, of the kind of preparedness that appeals to us.

Just as England can be said to be a nation of shop-keepers, so we can be called a nation of manufacturers. Any kind of preparedness, to be lasting with us, must have a peace value as well as a war value; otherwise, it can never endure through the coming years of peace and through changing political administrations. We will never maintain a big army; therefore, we must make a small army efficient, and must be ready at a moment's notice to increase it and supply it with arms and munitions without limit. We have here in America greater manufacturing and producing resources than any other two countries of the world. With these resources once thoroughly organized for the government service we need have little fear of attack. No other country can hope to compete with us in the quantity manufacture of the munitions of war.

TREMENDOUS GAGE REQUIREMENTS

Four rifles are needed for every soldier in the trenches. Recent testimony before the Congressional Committee on Military Affairs showed that 120 gages are needed in the manufacture of the bolt mechanism alone of a rifle, that many sets of gages are needed in each shop, and that many of these gages must be replaced after each ten to fifteen thousand gagings. It also developed that some 250

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gages are needed in the manufacture of a shell fuse; also that it would require the available gage-making facilities of this country five years properly to prepare American manufacturers to produce shells in such numbers as are being used daily on any one of several of the fighting fronts in Europe. Another interesting phase of this subject is that the large gage-making companies are all in New England, two of them within gun range of deep water.

It has been estimated that the money value of the gages necessary for fitting out American manufacturers to produce 200,000 shells a day throughout the range of the sizes used—some seventeen—is between seventeen and twenty million dollars.

The Conference Bill, which has just been signed by the President, empowers the Secretary of War to purchase wherever procurable and without the red-tape of a low-bid restriction the special tools, jigs and gages for supplying munitions to the army determined on by the General Staff as necessary for the defense of this country.

American manufacturers have found it almost impossible to procure the gages necessary for making munitions. The foreign governments, with the exception of Germany, were not prepared to supply gages and American manufacturers had to learn all of the details and kinks in manufacturing them. They have had to make up master parts, get these parts approved, and then design and make the gages, jigs and special tools for quantity production.

VALUE OF MACHINE GUNS

Germany had, it is said, between 50,000 and 60,000 machine guns at the outbreak of the war. England is said to have had less than 1000, and at the rate England was getting them six months after the war began it would have taken her nearly thirty years to make up Germany's lead, to say nothing of the number Germany is turning out per day or per month.

It is estimated roughly that anywhere along the fighting line two men with a machine gun are worth a thousand men armed with rifles. That accounts for the ability Germany has shown in shifting men from one front to another. She has found it necessary to leave only a thin line of troops equipped with machine guns in order to hold in check practically any advance upon the part of the Allies.

Another interesting phase of the subject is that new orders placed in this country for machine guns call for six extra barrels for each gun. They are now calling for hardened steel shields—a small conical shield—at the muzzle of the machine gun, for preventing the plugging of the barrel by the sharpshooters in trench often only a few yards away. Ten limbers are now specified for delivery with each field gun for foreign service. I am told that the American manufacturers have more trouble to-day in supplying the limbers than the guns themselves.

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STANDARDIZATION OF MUNITIONS NEEDED

The making of munitions is a parts-making business. In case this country ever goes to war with a first-class foreign power, every one of our plants having machine-tool or producing equipment will be mustered into government service. Parts made in Denver, Col., must be made to fit parts made in California, in Ohio or in Connecticut.

There are some thirty-five thousand items in the list of government supplies. We have no more reason to believe that the drawings and specifications developed in our own military departments are more suitable for quantity and economical production than are the similar specifications of foreign powers. Many changes must be made to facilitate production; gun sizes should be decreased in number and standardization must take place all along the line. There is at least one year's work for a commission of practical quantity-producers in the standardization and correction of the munitions specifications and drawings in our departments in Washington, before we even start to prepare this country to defend itself.

Theoretically, the Naval Consulting Board is an advisory body, a consulting and advising arm of the Navy Department. But its Committee on Industrial Preparedness has far exceeded the limits of the advisory function for which the Board was originally created. We have realized that advice is easy to give, but seldom taken; and so, taking the bull by the horns, we have gone out during the past five months to make our Committee of the greatest practical use to the country, as well as to the Navy and War Departments. Under the changed conditions of warfare, industry must be given its due place. In war as now waged, as a matter of fact, there are three "graces": the Army, the Navy, and Industry, and the greatest of these is Industry.

The making of munitions of war is a new art. It is an art we must teach our manufacturers as a fundamental step in any kind of preparedness. The average American factory has no more reason to assume that it can turn out modern military rifles or time fuses than that it can turn out Swiss watches. The superintendent of one of our famous sporting gun factories remarked a short time ago that his company had no more business taking a certain foreign rifle order than Caruso had. This statement was made after a year's struggle to make rifles that would pass inspection, and after fifty years in the gun business. The statement is made on the best of authority that with all the talk about American-made munitions, not a single American shell has yet been fired from a French gun, and no American rifles are yet in active service at the front.

WORK OF INDUSTRIAL PREPAREDNESS COMMITTEE

The campaign of the Committee on Industrial Preparedness of the Naval Consulting Board has been based on three steps: first, to

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find out what the country can produce in the way of army and navy supplies in case of need. An industrial inventory is in process for this purpose. Several States have practically finished their inventory and the reports are already in the New York office. In Texas, for instance, only three companies refused to fill out the forms. Practically all the information desired from them was obtained by the field aids anyway.

Second, teach every manufacturer in the United States to make for the government that one part or thing for which his equipment is best fitted. Small annual educational orders must be given to our manufacturers. These orders will not be of sufficient size to interfere in any way with regular commercial products. Suppose, for instance, that a motor-car plant is given a standing yearly order for only 10 shells of a certain size. The Purchasing Department will learn its sources of material supply. The Manufacturing Department will learn manufacturing methods and heat treatment. The Inspection Department will be given the required information as to governmental standards of inspection. The Engineering Department will have in its files uptodate specifications and drawings covering the work in detail and the special tools, jigs and gages for shop use in quantity production. The Shipping Department will learn methods of packing and routing of shipments. Last, but not least, the business end of the institution will be brought more closely in touch with governmental affairs and with governmental methods of business. The benefits of this closer business contact will accrue to both the government and the manufacturer.

Third, so enroll the skilled labor of the country that it cannot enlist in the Army or Navy in case of war.

The Committee on Industrial Preparedness has, in this work it has undertaken, the backing of the President and of the Secretaries of War and of the Navy. It has been aided by many other of the departments of the Government. The recent vote of the 980 Chambers of Commerce, members of the Chamber of Commerce of the United States, was practically unanimous in favor of the Committee's activity.

A board of directors of five has charge of the industrial inventory in each State—including Alaska—under direction of this Committee of the Naval Consulting Board.

Under these State Boards are serving, at the direct request of the President of the United States, more than 30,000 members of the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Society of Mechanical Engineers, the American Institute of Electrical Engineers, and the American Chemical Society.

I have been asked many times why the Society of Automobile Engineers was omitted from this list. There were two or three reasons. In the first place the Society is not national in a geographical sense. We are not in a position to appoint directors and furnish a field-aid

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corps in many States of the Union. Moreover, the work of this organization is a more specialized one and will be restricted to a more definite field. I will touch on this later.

INDUSTRIAL INVENTORY IN PROGRESS

The Industrial Inventory forms are now in the hands of field aids and manufacturers. Thousands of them are already coming into the New York office. The results of the inventory will be tabulated and delivered to the War and Navy Departments in due time.

At this point let me read an extract from an article in the May 3rd issue of one of the English technical papers:

"Ever since the invasion of Belgium the United States Government has realized the danger and humiliation which result from the pose as the guardian of the rights of neutrals and the ideals of humanity, and that to be unable to make effective protest against their destruction is perhaps the most invidious position in which a wealthy and proud nation can be placed. The moral of the situation has not been lost on the United States Government. Apart from the extensive naval program which has been adopted, steps have been taken to mobilize the industrial resources of the country in the event of war. The Naval Consulting Board has undertaken to make an industrial census with a view to arriving at the physical capacity of the plants and their fitness and reliability for specific classes of output.

"The ultimate object in view is that, in the event of war, orders for munitions and auxiliary equipment may be allocated right away in such a manner as to employ the industrial resources of the country in the most effective manner. Great Britain, with its usual lack of foresight, had to improvise all that machinery after the war. America has undoubtedly been impressed with the importance of preparedness on the engineering as well as the purely military side of the war. But we may note an even more significant feature of this American move. The census is taken through the agency of engineers, working voluntarily; and its results are to be crystallized by private individuals who have been chosen for exceptional experience in commercial statistical work. The British method is to create an army of incompetent officials, presided over by a typical civil servant. America's compliment to its engineers and commercial men will bring enthusiasm and efficiency in a degree which no government department could ever attain."

NEED FOR EDUCATIONAL MUNITIONS WORK

In the event of a defensive war waged against aggression upon the part of any one of the several first-class powers, we will need every ounce of national resource, both in industry and in men. It is

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clear, therefore, that when a manufacturer cannot or will not serve the needs of the Government, in support of either the fighting line or the civilian population, in accordance with the classification of his plant for the national service, there will remain no other alternative than to use the plant's labor and equipment in other fields.

A country-wide distribution of educational munition work through a careful prearrangement in time of peace will in the event of war have the greatest possible stabilizing influence upon our economic system. Labor will be kept employed and the suffering usually attending conditions of war will be reduced to a minimum. The manufacturer will no longer be uncertain as to the course which he is to follow in such an emergency. He will hold his labor organization together. Time loss and money loss will be lessened or entirely avoided. In case of necessity, only a telegram should be needed from the War Department to a plant with the one word "Proceed." The program within that plant should be so thoroughly understood that there would be no doubt as to the meaning of the telegram. Contrast that, if you like, with the months of argument, of experimentation and of delay which would result now were war declared and that plant suddenly called into government service. The government has no machinery at present for teaching munitions work, and thus the benefits of such peace-time educational work will be mutual.

WOMAN'S PART IN WAR

There is no longer any question as to woman's part in war. Three hundred thousand English women are at work in the mills and machine shops making war supplies. And these women are drawn, not from the working class but from the class known as the leisure class. Other thousands are nursing, acting as railway conductors, and driving cabs. A Frenchwoman said to me a few days ago, "France has had two saviors—its women and its seventy-fives."

MOTOR VEHICLES SAVED FRANCE

But I think we may add a third. The motor vehicle saved Paris and France itself in the battle of the Marne. It has saved France a dozen times since then. The motor vehicle has made possible the strategic moves of the European conflict. The motor vehicle has been the mainstay of transportation behind every fighting front. The German expedition into Russia would have been impossible without it.

In our country on Jan. 1, 1916, 2,400,000 motor vehicles were in service. In all the rest of the world put together there were less than 800,000; in France less than 100,000. Since Jan. 1, 400,000 new cars have gone into service in the United States. By Jan. 1 next, 3,400,000 will be in active service—more than four times the number in service in all the rest of the world put together.

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NATIONAL PLAN FOR TRANSPORT MOBILIZATION

But to make these motor vehicles available for the governmental service in case of emergency, a national plan of mobilization must be worked out and put to practical test in time of peace. Here, as in every other phase of preparedness, we can progress with certainty only through prearrangement. A national plan of motor transport must be evolved for this emergency service. No such plan now exists. A committee of five has already been selected by the Council of the Society of Automobile Engineers for cooperation with the War College General Staff. The National Automobile Chamber of Commerce and the American Automobile Association have also selected committees. A similar step has been taken by the American Railway Association, thus bringing to the aid of the General Staff the best brains in the country on rail and motor transportation matters.

Let me read you an extract from the statutes which explains the inability of the departments in Washington to cooperate with a civilian body in any line of work such as the mobilization of motor vehicles for transport service. The provision states:

"Nor shall any department or any officer of the government accept voluntary service for the government or employ personal service in excess of that authorized by law except in case of sudden emergency involving the loss of human life or the destruction of property."

In short, the War Department not only has no power to call but is prohibited by statute from calling on this engineering body for cooperation in work relating to any phase of the activities looking toward defense.

But the passage of the Conference Bill will make it possible to organize and officer such transportation units as may be created through joint planning and agreement. The officers and men for this transport service must come from the motor-car industry and from the railways.

If the railways are taken into government service, as they must be in the event of war or of the imminence of war, officers will be assigned from the War Department to cooperate in the management of the lines, but at the same time the operating heads of the various divisions of the railways will be given an official ranking as officers of the department. The same thing must happen in the motor transport. The officers and the personnel of the entire motor-transport system must be drawn from the motor-car industry; and logically, I should say, the officers must be drawn from the membership of the Society of Automobile Engineers. All of this must be done in time of peace. It will be impossible to do it under the chaotic conditions following a declaration of war.

PROVISIONS OF CONFERENCE BILL

There are one or two other provisions of the Conference Bill to which I would like to call your attention. (This is the combination

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of the Chamberlain and the Hay Bills, finally agreed upon in conference.) I will read you the section providing for officers not only in transportation lines but in any other line in which governmental cooperation is deemed necessary.

Under Section 37, "The President alone shall be authorized to appoint and commission as reserve officers in the various sections of the Officers Reserve Corps, in all grades up to and including that of major, such citizens as, upon examination prescribed by the President, shall be found physically, mentally and morally qualified to hold such commissions."

I think most of the members of the Society of Automobile Engineers might be considered to fill these requirements.

The bill also provides that the low-bid rule shall not necessarily maintain. The War Department now has the power to go out and place educational orders with manufacturers in every part of the country and in every industry, and teach them how to make some one thing for the service. This is entirely in keeping with the program for which the Naval Consulting Board has been working during the last six months.

PREPAREDNESS TASK FOR ENGINEERS

And now, one final word. Do not depend on Congress to save this country or prepare it for a proper defense. The lawyer and the politician have no place in shaping the industrial foundation which must become at once our insurance against war and our assurance of victory in case war is forced upon us. The problems are industrial problems and must be handled by those men skilled in quantity manufacturing and in the organization of great industries. I do not mean to reflect in any way upon Congress or criticize the action of that body. But our congressmen cannot appreciate the intricate industrial machinery, which is our basic strength for defense. Months have been devoted by these gentlemen to discussing the Army and Navy. As to the organization of our industrial resources, we have had little either in discussion or in action. This problem must be solved by our engineers and manufacturers—and the sooner we realize it and begin to study it the better. Let us work calmly and efficiently in time of peace and through careful prearrangement avoid the disasters that threaten the other great nations of the world.

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APPLICATIONS APPROVED BY THE COUNCIL SINCE JUNE, 1916, MEETING

Members	78
Associates	56
Juniors	30
Affiliates	2
Student Enrollments	15
Total	181

November, 1916.

ATTENDANCE AT 1916 COUNCIL MEETINGS

	Possible	Actual
Russell Huff, President	9	9
Eugene S. Foljambe, First Vice-president	9	8
Robert H. Combs, Second Vice-president	9	4
Herbert Chase, Treasurer	9	9
William P. Kennedy, Councilor	9	5
C. B. Rose, Councilor	9	2
David Beecroft, Councilor	9	5
George W. Dunham, Councilor	9	7
John G. Utz, Councilor	9	7
*Jesse G. Vincent, Councilor	4	2
†Henry M. Leland, Past-president	7	0
§Charles W. McKinley	2	1
W. H. VanDervoort, Past-president	9	2
A. Ludlow Clayden, Chairman Standards Committee	7	6
H. M. Swetland, Chairman Finance Committee	9	0
Coker F. Clarkson, Secretary	9	9

*Succeeded Councilor Hall.
†Resigned September, 1916.
§Succeeded Past-president Leland.

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TREASURER'S REPORT

The treasurer's report for the twelve months ending Sept. 30, 1916 was submitted. On motion, duly seconded, this report (which follows) was accepted and ordered placed on file.

BALANCE ON HAND, OCT. 1, 1915:

Certificate of Deposit	$7,000.00	
Checking Account	1,487.94	
Cash in Office	45.00	
		$8,532.94

Receipts

Initiations	$8,360.00	
Dues	27,361.00	
Pin, Binder and Folder Sales	1,725.77	
Publication and Other Receipts	62,094.17	
	99,540.94	
Less Discount	440.76	
Total Receipts		$99,100.18
		$107,633.12

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Total Receipts Brought Forward		$107,633.12
Disbursements		
Salaries	$16,312.69	
Auditing	300.00	
Pin, Binder and Folder Stock	898.79	
Postage	2,126.96	
Publication Expense	9,504.55	
Rent	2,692.20	
Council & Committee	35.72	
Meetings Expense	24,473.53	
Telephone & Telegraph	553.19	
Carfare	16.60	
Office Expense	222.42	
Express	792.87	
Stationery & Supplies	1,265.19	
Furniture & Fixture	627.20	
Standards Committee	9,785.20	
Convention Reporting	373.25	
Library Expense	56.15	
Investment	9,694.17	
Unclassified	5,387.46	
	85,118.14	
Less Discount	370.39	
Total Disbursements		$84,747.75
		$22,885.37

BALANCE ON HAND, SEPT 30, 1916:

Certificate of Deposit	$20,000.00	
Savings Account	2,000.00	
Checking Account	860.37	
Cash in Office	25.00	
		$22,885.37

I hereby certify that the above statement is correct.

H. D. DABNEY,
Public Accountant.

October 14, 1916.

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FACTORS OF SAFETY

By Russell Huff

(President of the Society)

Abstract

The author has selected fourteen automobiles on which to make a study of the factors of safety used in their design. He considers specifically the front axles, front-wheel spindles, propeller-shafts, clutch-shafts, transmission drive-shafts and rear-axle drive-shafts. The method of calculating the stresses is outlined; compositions of the steels used are given; and complete data are presented showing the factors of safety of the various parts, together with the intermediate figures used in obtaining the factors.

Formulas for guiding the engineer in designing the vital parts of automobiles or motor trucks have been published from time to time, but data on the actual conditions of design as they exist in the average automobile of to-day are not available in a convenient, condensed form. The author, having made an intimate study of a number of different models of successful cars, presents in this paper comparative data on designs, materials, stresses and factors of safety, as found to exist in the several cars examined, in the front axle, front-wheel spindle, rear-axle drive-shaft, clutch-shaft, transmission drive-shaft and propeller-shaft. The fact that so much of automobile engineering practice is based on experience, rather than on theory, has prompted the author to make the study of the stresses and factors of safety given in this paper in order to see if there is any approximate uniformity in the designs commonly used by successful engineers and manufacturers.

The data submitted have been collected from a variety of cars ranging from a small 1900-lb. four-cylinder car to a 3-ton truck. Included in the list are cars with front axles of I-beam and tubular section, engines of four, six and twelve cylinders, transmissions located on the rear axle and forward in the frame, and rear axles of the semi-, three-quarter- and full-floating construction. Two 3-ton trucks, one of the chain-driven and the other of the worm-driven type, are covered in the list. The parts relating to the data submitted have been carefully measured and, with few exceptions, analyzed chemically.

Actual physical tests could not conveniently be made on many of the parts examined, but as physical properties of different kinds of

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TABLE I—GASOLINE ENGINE DIMENSIONS AND CHARACTERISTICS

Car No.	Car Weight Loaded, Lb.	Car Wheel-base, In.	Engine Bore, In.	Engine Stroke, In.	No. Cyl.	Piston Disp., Cu. In.	Estimated Maximum Torque, Lb. In.
1	3,315	118	4⅛	4½	4	240.5	1550
2	3,175	112	3¾	5	4	220.9	1450
3	2,523	100	3⅝	5	4	206.4	1350
4	2,828	105	3⅛	5	4	153.4	1010
5	4,360	116	3¼	4½	6	224.0	1510
6	2,932	110	3⅞	4½	4	212.3	1400
7	2,850	110	2 13/16	4¾	6	177.0	1160
8	2,500	105	3¼	5	4	165.9	1095
9	5,200	129	5	5½	4	431.9	2800
10	5,445	139	4½	5½	6	524.0	3360
11	13,870	126	4½	5½	4	349.9	2240
12	5,937	144	4	5½	6	414.7	2700
13	5,635	125	3	5	12	424.0	2750
14	14,070	156	4½	5½	4	349.9	2240

steels, with certain heat treatments, have been so well established by metallurgists, the author has assumed, in submitting the figures herein, that each manufacturer is now actually heat-treating the different steel parts of his automobiles according to the most up-to-date methods. Such parts therefore should show physical properties equal to those given in the S. A. E. Standard specifications.

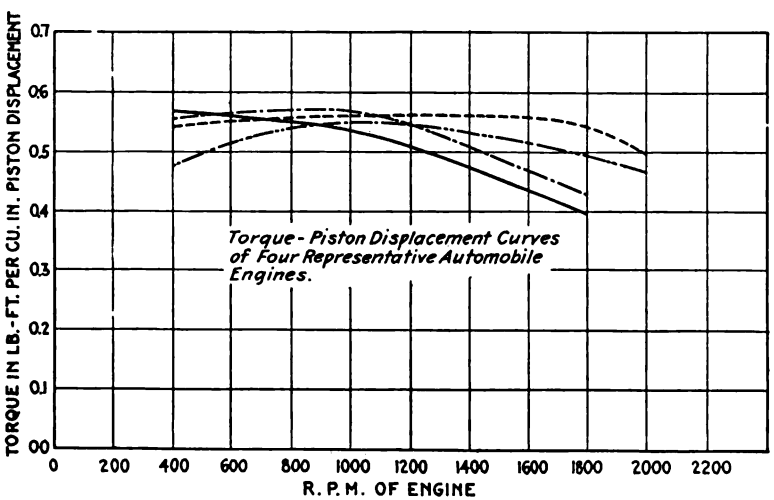

FIG. 1—TORQUE CURVES OF FOUR AUTOMOBILE ENGINES

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In the S. A. E. Handbook under specifications for steels, curves are presented showing the variation in the elastic limits to be expected from different steels, depending on the Brinell hardness. Inasmuch as these curves are drawn for the minimum average of results obtained and cover only plain carbon steels, the author found it necessary, in order to preserve uniformity throughout this paper, to select values from similar curves in his possession, covering plain carbon as well as alloy steels and giving average results from extended tests. A hardness numeral of 210 Brinell has been adopted for all except case-hardened parts, because machining operations after heat treatment to this hardness value are satisfactory. The figures on expected elastic limits submitted in the different tables throughout this paper are based on the assumption that the steel has been heat treated to give the same hardness.

In the study of front axles the actual dead load resulting from a full complement of passengers, full gasoline, oil and water systems

TABLE II—COMPOSITION (IN PER CENT) OF FRONT-AXLE STEELS

Car No.	S. A. E. Steel No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium
1	1045	0.417	0.82	0.020	0.038	0	0	0
2	1035	0.305	0.82	0.018	0.031	0	0	0
3	1035	0.334	0.78	0.016	0.035	0	0	0
4	1025	0.250	0.46	0.014	0.021	0	0	0
5	1035	0.321	0.60	0.018	0.039	0	0	0
6	6125	0.280	0.70	0.018	0.035	0	0.90	0.18
7	1035	0.349	0.72	0.018	0.035	0	0	0
8	1035	0.364	0.48	0.013	0.045	0	0	0
9	2330	0.250	0.50	0.020	0.036	3.50	0	0
10	3140	0.400	0.60	0.010	0.030	1.20	0.35	0
11	1025	0.250	0.50	0.015	0.035	0	0	0
12	3140	0.400	0.60	0.010	0.030	1.20	0.35	0
13	3140	0.400	0.60	0.010	0.030	1.20	0.35	0
14	3140	0.400	0.60	0.010	0.030	1.20	0.35	0

and spare tire equipment was used in making the calculations. In the study of the power-transmission elements, such as the clutch-shaft and rear-axle drive-shaft, the maximum engine torque, direct or geared-up, irrespective of friction or gear losses, was used in all the calculations. The lowest possible ratio was used in the consideration of all elements affected by a speed reduction through gears. Many of the engines were tested on the dynamometer to determine a fair average torque for each engine. By reducing the data thus obtained to pound-feet torque per cubic inch piston displacement, the figure of 0.55 was obtained and has been used throughout in this paper as being a reasonable average for all engines in the cars examined. (Fig. 1 shows torque curves from four representative engines.) Table I gives the loaded car weights, wheelbases, engine data, and estimated maximum engine torque for all the cars under consideration.

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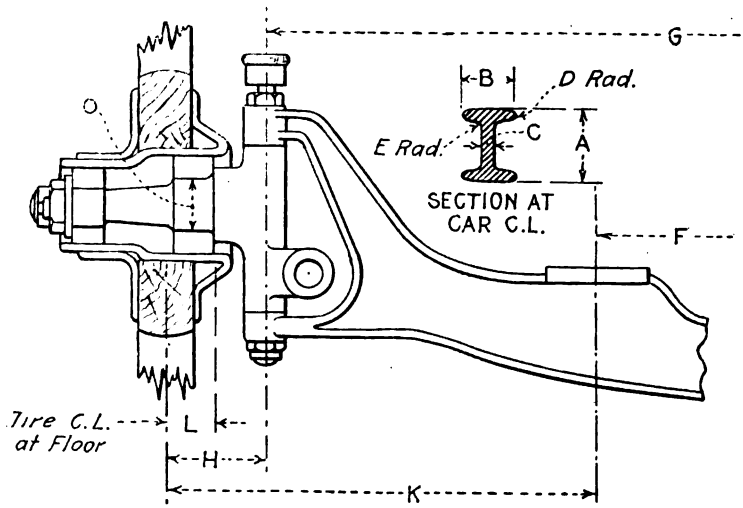
FRONT AXLES

Of the fourteen front axles examined, one had the tubular, one the solid rectangular and all the others the I-beam section. Chemical analyses of the axle materials show that nickel, chrome vanadium, chrome nickel and plain carbon steels are used.

In studying the stresses to which front axles are subjected, it is customary to consider only the bending moments in the vertical plane, due to the dead weight on the axle. There are of course unusual

TABLE III—DIMENSIONS OF FRONT AXLES AT CENTER SECTION



Car No.	A (=2×c)	B	C	D	E	Area in Sq. In.
1	2 3/16	1 7/16	1/4	1/8	5/16	1.08
2	2 3/8	1 3/4	3/16	3/32	3/8	1.44
3	1 15/16	1 1/4	1/4	3/32	3/8	0.96
4	2	1 7/16	1/4	3/32	7/16	1.12
5	2 1/8	1 1/2	1/4	3/32	3/8	1.00
6	2	1 3/8	9/32	7/64	5/16	1.20
7	1 3/4	1 1/4	1/4	3/32	3/8	1.00
8	1 7/8	1 1/2	1/4	3/32	1/16	0.90
9	Tubular	Axle—I.	D. 2—O.	D. 2 1/2 in.		1.75
10	2 13/16	2	1/2	3/16	.....	2.53
11	3 7/16	2 1/4	.....	.....	.....	7.72
12	2 1/4	1 7/8	7/16	5/32	.....	1.88
13	2 5/16	1 7/8	7/16	5/32	.....	1.86
14	3 1/4	2 1/2	7/16	3/16	.....	2.95

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TABLE IV—FRONT AXLE DIMENSIONS AND STRESSES

Car No.	Weight on Frt. Axle, Lb.	F*	G*	H*	K*	Bending Moment on Frt. Axle, Lb. In.	Moment of Inertia of Front Axle	Section Modulus Relative to		Ratio Vert. to Hor. Sect. Mod., Per Cent	Max. Fiber Stress, Lb. per Sq. In.	S.A.E. Steel No.	Expected Elastic Limit, Lb. per Sq. In.	Factor of Safety
								Horizontal Axis	Vertical Axis					
1	1375	31½	51¾	2⅛	12¼	8,425	0.777	0.710	0.139	19.6	11.950	1045	70,000	5.9
2	1335	28½	52¼	2½	14⅜	9,600	1.190	1.005	0.229	22.6	9,600	1035	70,000	7.3
3	1065	27 1/16	51⅞	2½	14 29/32	7,950	0.510	0.526	0.157	29.8	15,100	1035	70,000	4.6
4	1300	28	51⅜	2 5/16	14	9,100	0.672	0.672	0.167	24.8	13,550	1025	55,000	4.0
5	1550	28½	50¾	2⅝	13¾	10,650	0.667	0.627	0.157	125.1	17,000	1035	70,000	4.1
6	1235	26¼	50¾	3 3/64	15 19/64	9,430	0.897	0.897	0.178	19.8	10,500	6125	90,000	8.6
7	1185	26⅜	51½	2⅜	14 15/16	8,850	0.460	0.526	0.128	24.3	17,000	1035	70,000	4.1
8	1255	26¾	51¼	2 3/16	14 7/16	9,020	0.475	0.506	0.140	27.7	17,900	1035	70,000	3.9
9	2085	29⅜	48⅝	3 11/16	13 3/16	13,800	1.130	0.906	0.906	100.0	15,250	2330	90,000	5.9
10	2300	26⅜	50	3	14 11/16	16,860	1.892	1.345	0.525	39.0	12,500	3140	85,000	6.8
11	3870	27	54½	4 11/16	18 7/16	35,700	7.610	4.310	2.900	67.0	8,050	1025	55,000	6.9
12	2520	26⅝	50 13/16	2 19/32	14 11/16	18,480	1.430	1.270	0.489	38.5	14,520	3140	85,000	5.9
13	2400	26⅝	50 13/16	2 19/32	14 11/16	17,600	1.490	1.290	0.494	38.3	13,650	3140	85,000	6.2
14	4020	36	62½	4½	17¾	35,700	4.680	2.880	0.878	30.5	12,400	3140	85,000	6.9
									Average	...31.0			Average	..5.8

* drawing of axle in Table III for key to reference letters.

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loads to be sustained by the axle in both the vertical and horizontal planes when the wheels drop into chuck-holes or strike obstacles in the road at high speed. These special stresses cannot be determined in a practical manner because the conditions are variable. However, accumulated experience has taught the engineer that, if a certain factor of safety based upon the vertical stresses due to the straight dead load is provided, the axle will stand up successfully. Experience has also taught the engineer that certain proportions between the vertical and the horizontal resisting moment of the axle section must be maintained.

Tables II, III and IV cover the essential data relating to front axles. Table II gives the chemical analyses and the equivalent S. A. E. steels; Table III the important dimensions of the center sections of all the

TABLE V—COMPOSITION (IN PER CENT) OF FRONT-AXLE SPINDLE STEELS

Car No.	S. A. E. Steel No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium
1	1035	0.386	0.67	0.017	0.027	0	0	0
2	5140	0.335	0.71	0.014	0.030	0	0.92	0
3	3140	0.417	0.46	0.031	0.025	1.15	0.58	0
4	1025	0.250	0.46	0.014	0.021	0	0	0
5	1045	0.435	0.66	0.014	0.033	0	0	0
6	6125	0.280	0.70	0.018	0.035	0	0.90	0.18
7	1035	0.300	0.68	0.010	0.035	0	0	0
8	3335	0.370	0.53	0.040	0.036	3.02	0.57	0
9	1035	0.400	0.55	0.015	0.025	0	0	0
10	1035	0.400	0.55	0.015	0.025	0	0	0
11	1035	0.400	0.55	0.015	0.025	0	0	0
12	3140	0.400	0.60	0.010	0.030	1.20	0.55	0
13	3140	0.400	0.60	0.010	0.030	1.20	0.55	0
14	3140	0.400	0.60	0.010	0.030	1.20	0.55	0

axles; and Table IV the data on the vertical bending moments, horizontal moments of inerta, fiber stresses, expected elastic limits of the steels used and factors of safety found in the axles.

The moments of inertia of front axle sections in Table IV were obtained by the graphical method in order to get them exactly, instead of by the approximate method commonly used in estimating moments of inertia of irregular sections.

To obtain the one most interesting value of all, namely, factor of safety, computations were made as follows: Taking the case of Car No. 1, we have in Table IV the weight on the front axle as 1375 lb. The weight then on each spring-pad would be one-half of this, or 687.5 lb. The bending moment resulting from this load would be this weight times the lever arm (the distance between the center of the front wheel and the center of the front spring), in this case 12¼ in.,

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and which, when multiplied by the weight, gives a bending moment of 8425 lb. in. This is the maximum bending moment on the axle and is the same at both spring-pad centers and at any point between these centers.

It is the author's observation that most front axles have a slightly weaker section at the center than at any other point, and therefore, in order to simplify the tables and calculations, measurements at the center only were used.

The maximum fiber stress was obtained by the common flexure formula: $M = sI \div c$, in which M is the bending moment, s the fiber stress, I the moment of inertia and c the distance from the center line of gravity of the section to the extreme fiber. In this formula the

TABLE VI—FRONT-AXLE SPINDLE DIMENSIONS AND STRESSES

Car No.	Load on Front Axle, Lb.	L*	O* (=2×c)	Spindle Bending Moment, Lb. In.	Moment of Inertia of Spindle at Bearing Shoulder	Max. Fiber Stress, Lb. per Sq. In.	S.A.E. Steel No.	Expected Elastic Limit, Lb. per Sq. In.	Factor of Safety
1	1375	1¼	1⅜	860	0.176	3360	1035	70,000	20.9
2	1335	1¼	15/16	848	0.146	3810	5140	80,000	21.0
3	1065	1 3/16	1 3/16	634	0 098	3840	3140	85,000	22.2
4	1300	1¼	1⅛	812	0.077	5940	1025	55,000	9.3
5	1550	1⅝	1 7/16	1260	0.209	4325	1045	70,000	16.2
6	1235	1 13/32	15/16	870	0.146	3930	6125	90,000	22.9
7	1185	⅞	1¼	517	0.120	2690	1035	70,000	26.1
8	1255	1⅜	1⅛	865	0.077	6300	3335	90,000	14.3
9	2085	2¼	1 31/32	2350	0.735	3140	1035	70,000	22.3
10	2300	1⅛	1 31/32	1294	0.735	1735	1035	70,000	40.4
11	3870	2¾	2⅝	5320	2.34	2980	1035	70,000	23.5
12	2520	1	1 13/16	1260	0.532	2150	3140	85,000	39.6
13	2400	1	1 13/16	1200	0.532	2045	3140	85,000	41.6
14	4020	1 9/16	2 0/16	3145	2.13	1895	3140	85,000	44.8
								Average	...26.1

*See Table III for key to reference letters.

values for M, I and c are known, and solving s gives us the maximum unit fiber stress in the steel. Illustrating this with Car No. 1; $M =$ 8425 lb. in., $I = 0.777$ and $c = 1\ 3/32$ in. Substituting these values in the formula, we have

$$8425 = \frac{s \times 0.777}{1.09375} \quad \text{or} \quad s = \frac{8425 \times 1.09375}{0.777} = 11{,}950 \text{ lb. per sq. in.}$$

The factor of safety is obtained by dividing the expected elastic limit of the steel by the unit fiber stress. In the axle for Car No. 1

$$FS = \frac{70{,}000}{11{,}950} = 5.9$$

The average of all the factors of safety for front axles, as shown in the right-hand column of Table IV, is 5.8, and, since all of the

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cars examined in this study are recognized as successful, it is reasonable to believe that any front axle having this factor of safety in the vertical plane will successfully carry the dead-weight load and resist all usual shocks resulting from the wheels striking obstacles in the road.

As previously stated, practical experience extending over a period of years has taught the engineer that certain proportions between the vertical and horizontal resisting moments must be maintained. In order to find an average of this proportion, the modulus of each axle section, relative to both horizontal and vertical axes, has been determined. These values are given in Table IV, as is also the ratio (in per cent) of the vertical to horizontal modulus. The average of this ratio, with Car No. 9 omitted, is 31 per cent, showing the engineer

TABLE VII—COMPOSITION (IN PER CENT) OF CLUTCH-SHAFT STEELS

Car No.	S. A. E. Steel No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium
1	1020	0.21	0.34	0.009	0.032	0	0	0
2	Malleable Iron	.....	.....	.....	.....	.....	.....	.....
3	3125	0.25	0.51	0.011	0.023	1.25	0.61	0
4	1020	0.19	0.32	0.013	0.042	0	0	0
5	2330	0.25	0.63	0.015	0.028	3.01	0.20	0
6	6115	0.12	0.33	0.030	0.025	0	0.28	0.14
7	3540	0.44	0.51	0.032	0.036	2.85	1.01	0
8	6115	0.15	0.32	0.031	0.026	0	0.29	0.13
9	Special	0.43	0.45	0.025	0.040	4.5	0	0
10	Special	0.43	0.45	0.025	0.040	4.5	0	0
11	Special	0.43	0.45	0.025	0.040	4.5	0	0
12	Special	0.43	0.45	0.025	0.040	4.5	0	0
13	Special	0.43	0.45	0.025	0.040	4.5	0	0
14	Special	0.43	0.45	0.025	0.040	4.5	0	0

has found it practicable to make front axles about three times stronger in the vertical than in the horizontal plane.

FRONT AXLE SPINDLES

Examination of steering spindles shows a marked similarity in construction. Later models have the spoke center-line as near the inner bearing shoulder as practicable. This is good design, because it reduces the bending moment produced on the spindle by the vertical load and by horizontal shocks. It also reduces the friction in the knuckle thrust and pivot bearings, thus tending to easier steering. A third advantage lies in the shorter turning radius obtained by this construction. The value of ample fillets at the spindle-bearing shoulder is recognized by all designers, as shown by the large fillets found in the cars examined.

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In order not to make this study too long, the only analysis submitted is that of the stresses and factors of safety found at the spindle-bearing shoulder. In each case the steel was analyzed in order to determine its equivalent S. A. E. number and the elastic limit expected. These analyses, with the equivalent S. A. E. steel numbers, are given in detail in Table V. The data on spindle sizes, loads, bending moments and stresses, are given in a condensed form in Table VI.

The same formula used for determining the unit fiber stress in the front axles, namely, $M = sI \div c$, was employed in obtaining the spindle fiber stresses given in Table VI. The factors of safety were also obtained in a manner similar to the method used in the front axle calculations.

TABLE VIII—CLUTCH-SHAFT DIMENSIONS AND STRESSES

Car No.	Max. Torque on Clutch Shaft, Lb. In	Length, In.	Least Diameter, In.	Polar Moment of Inertia, Weakest Section	Max. Fiber Stress, Lb. per Sq. In.	S.A.E. Steel No.	Expected Elastic Limit, Lb. per Sq. In.	Factor of Safety
1	1550	7½	1 3/32	0.125	7,000	1020	40,500	5.8
2	1450	3 5/16	2⅛	5.600	370	Mall. Iron	30,000	81.0
3	1350	7⅜	1½	0.187	5,420	3125	100,000	18.5
4	1010	8 29/32	1¼	0.240	2,630	1020	50,000	19.0
5	1510	11¼	1⅜	0.350	3,000	2330	71,000	23.6
6	1400	9 13/32	1⅛ sq.	0.221	4,250	6115	40,000	9.4
7	1160	7 5/16	1⅞	0.980	1,110	3540	155,000	139.5
8	1095	4⅛	1	0.194	3,700	6115	40,000	10.8
9	2800	16	1.1811	0.193	8,250	Special	100,000	12.1
10	3360	16	1.1811	0.193	10,280	Special	100,000	9.8
11	2240	16	1.1811	0.193	6,850	Special	100,000	14.6
12	2700	15 11/32	1.5750	0.604	3,500	Special	100,000	28.6
13	2750	13	1.3125	0.294	6,150	Special	100,000	16.3
14	2240	16	1.1811	0.193	6,850	Special	100,000	14.6
							Average...	28.8

Illustrating these computations in the case of Car No. 1, by substituting in the formula the values in the table for M, I, and c, we have the following:

$$860 = \frac{s \times 0.176}{0.6875}$$

$$\text{or} \quad s = \frac{860 \times 0.6875}{0.176} = 3360$$

Solving for the factor of safety, we have

$$FS = \frac{EL}{s} = \frac{70{,}000}{3360} = 20.9$$

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The factors of safety found in the spindles average much higher than those found in the axles. According to this it would be expected that less breakage would occur here than in the axle, but it has been the author's observation that this is not the case. The author believes this is due to extreme stresses concentrated at the shoulder, caused by the excessive side strains that occur when the wheel strikes a rut or curb in skidding. These strains are also high when steering through deep ruts that do not quite fit the front-wheel gage.

The most common form of side strain, but one that is not of high value, is the centrifugal-force side-thrust on the front wheels when turning corners in ordinary driving. This thrust can be calculated by knowing the weight on the tires, the speed of the car and the coefficient of friction between the tire and ground. The centrifugal force also introduces an added load upon the outside spindle, which can be computed when the height of the center of gravity of the car is known in addition to the data mentioned. The stresses from this cause are less than the vertical load bending moments, so that they have been ignored in this analysis.

CLUTCH-SHAFT

In studying the stresses and factors of safety in the clutch-shaft, main transmission shaft and propeller-shaft, only the weakest section against torsion was considered. As stated before, the value of 0.55 lb.ft. torque per cubic inch piston displacement was adopted as a standard for estimating the maximum torque from each engine on its respective transmission line. In Table VII the analysis and the equivalent S. A. E. steel specification for each clutch-shaft are submitted. In Table VIII are given data on the maximum torque, length, least diameter, polar moment of inertia, maximum fiber stress, expected elastic limit of the steel and factor of safety of each shaft.

The polar moments of inertia of the weakest section in each shaft have been carefully calculated. These values, together with the known values for torque and shaft diameters, make it an easy matter to comput the maximum fiber stress by the common torsion formula: torque $= sJ \div c$, in which s is fiber stress; J, polar moment of inertia; and c the distance of the most remote fiber from the axis in the cross section under consideration.

Again illustrating this with Car No. 1, we have a maximum engine torque of 1550 lb. in., a polar moment of inertia of 0.125 and a value for c of one-half the diameter of the weakest section, or 0.5468 in. Substituting these values in the formula, we have:

$$s = \frac{1550 \times 0.5468}{0.125} = 7000$$

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Using the usual formula for factor of safety, namely, $FS = EL \div s$, and substituting the values in Table VIII for E, L, and s we have

$$FS = \frac{40{,}500}{7000} = 5.8$$

Some extremely high factors of safety will be noted in this table. The engineers who designed these shafts either never figured them out carefully or else originally designed them for a low-grade steel and later on substituted a high-grade steel without reducing the dimensions accordingly. The average of the entire lot, which is 28.8, is much higher than the factors of safety found in front axles. It is

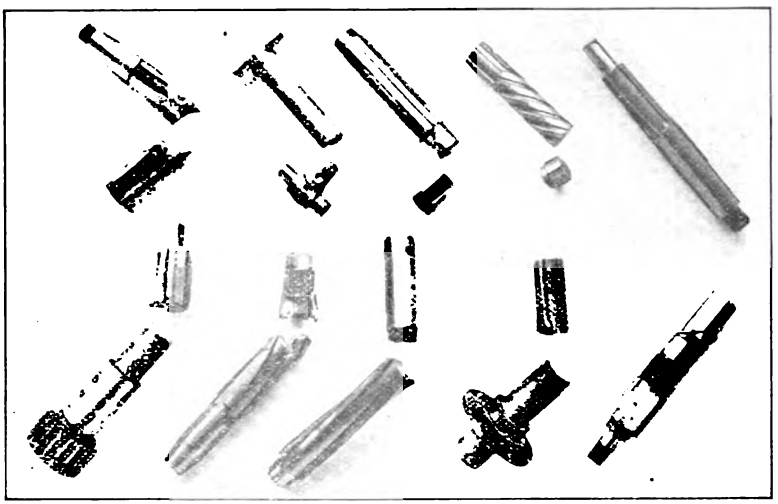

FIG. 2—TRANSMISSION DRIVE-SHAFTS TESTED TO DESTRUCTION

most unusual to hear of a broken clutch-shaft, and it would seem to the author that this one element in the average American car could be reduced in size and weight in many cars without risking any serious danger of breakage.

It is true that shocks on the shafts are high where savage clutches exist, but in the modern smooth-acting disk clutch and the well designed cone clutch, smooth action is the general thing, whereas a few years ago the reverse was common.

The study showed that a greater diversity of engineering practice exists in clutch-shafts than in any other element examined. This is probably caused by the innumerable combinations possible in clutch construction, where so many detail elements, such as bearings, shift-

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ing devices, universal joints, flywheels and pedals, are used. Of the fourteen clutches under consideration, seven were of the dry-plate and all the others of the cone type. The almost universal practice of attaching the transmission unit, together with the clutch housing, directly to the engine base, and the importance of making these two units as short and compact as possible, to reduce the unsupported overhang, is no doubt the cause of the prolific clutch-design combinations.

TRANSMISSION DRIVE-SHAFT

In examining the transmission drive-shafts, a much greater uniformity in engineering practice is observed. The four-spline shaft is rapidly superseding the old style square shaft. Many of the shafts examined were case-hardened while others were tempered by heat treatment. The physical properities of most of these hardened shafts

TABLE IX—COMPOSITION (IN PER CENT) OF TRANSMISSION-SHAFT STEELS

Car No.	S. A. E. Steel No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium
1	1045	0.492	0.42	0.038	0.035	0	0	0
2	3220	0.239	0.53	0.033	0.035	1.74	0.081	0
3	3130	0.293	0.38	0.009	0.027	1.15	0.058	0
4	1020	0.201	0.41	0.018	0.021	0	0	0
5	1045	0.487	0.65	0.015	0.032	0	0	0
6	6115	0.120	0.33	0.020	0.020	0	0.27	0.18
7	1030	0.295	0.42	0.014	0.053	0	0	0
8	1030	0.295	0.42	0.014	0.053	0	0	0
13	Special	0.430	0.45	0.025	0.040	4.5	0	0
9	Special	0.430	0.45	0.025	0.040	4.5	0	0
10	Special	0.430	0.45	0.025	0.040	4.5	0	0
11	Special	0.430	0.45	0.025	0.040	4.5	0	0
12	Special	0.430	0.45	0.025	0.040	4.5	0	0
14	Special	0.430	0.45	0.025	0.040	4.5	0	0

were obtained by testing them to destruction in a torsion machine. By this method the effect of the case-hardening and heat treatment on them was obtained. The shafts thus tested are shown in Fig. 2.

The figures on expected elastic limit submitted in Table X are based on the average results secured in these torsion tests. The data on chemical analyses of these shafts submitted in Table IX were secured from samples taken from the soft cores. The values for maximum fiber stress and factor of safety submitted in Table X were obtained by means of the same formulas used in the consideration of clutch-shafts.

In studying the value of factors of safety submitted, the author wishes to call attention to the fact that the first nine cars listed in Table X had transmissions designed with the constant mesh gears at the forward end, and the other five cars had transmissions with the constant mesh gears at the rear. The difference in these two gear

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layouts may explain why the factors of safety in the first nine are considerably lower than those in the remaining five. In the case of Car No. 9, the factor is unusually high since the shaft is exceptionally large for the size of the engine in the car. An average of the factors of safety of the first nine cars listed in Table X is 5.0, a value the author would feel justified in recommending as a safe figure to employ because all of these nine have a good reputation and so far as the author can find out have had no trouble with their transmission shafts. The factors of safety found in the last five cars average higher than practical experience would seem to indicate as necessary.

TABLE X—TRANSMISSION-SHAFT DIMENSIONS AND STRESSES

Car No.	Length of Shaft, In.	Least Diameter, In.	Lowest Gear Reduction	Maximum Torque, Lb. In.	Polar Moment of Inertia, Weakest Section	Max. Fiber Stress, Lb. per Sq. In.	S.A.E. Steel No.	Expected Elastic Limit, Lb. per Sq. In.	Factor of Safety
1	13 27/32	1¼	5:1	7,750	0.240	20,200	1045	85,000	4.2
2	12⅝	1¼	4.76:1	6,900	0.458	12,120	3220	60,000	5.0
3	8⅝	1 1/32	3.99:1	5,375	0.186	17,900	3130	80,000	4.5
4	10 11/16	1 3/16	3.24:1	3,275	0.280	8,400	1020	70,000	8.4
5	13¼	1⅛	4.3:1	6,500	0.238	18,000	1045	100,000	5.5
6	10 15/32	1⅛	4.66:1	6,520	0.205	21,850	6115	87,000	4.0
7	10 15/16	1 3/16	3.9:1	4,625	0.071	26,500	1030	115,000	4.4
8	9 9/16	1	3.63:1	3,970	0.098	20,300	1030	85,000	4.2
13	12 7/32	1 11/32	3.42:1	9,400	0.322	19,600	Special	100,000	5.1
9	12 23/32	1 5/16	1:1	2,800	0.248	7,400	Special	100,000	13.5
10	11 23/32	1 5/16	1:1	3,360	0.248	8,900	Special	100,000	11.2
11	12 23/32	1 5/16	1:1	2,240	0.248	5,950	Special	100,000	16.8
12	11 23/32	1 5/16	1:1	2,700	0.248	7,150	Special	100,000	14.0
14	11 13/64	1 5/16	1:1	2,240	0.248	5,950	Special	100,000	16.8
								Average	8.4

PROPELLER-SHAFTS

Propeller-shaft construction has had the critical study of the designing engineer during recent years because of the whipping and consequent buckling tendencies introduced by the higher engine speeds and the increase in shaft length made necessary by the longer wheelbases. It is not the object of this paper to go into the theoretical study of the strength of these shafts in relation to whipping tendencies, but rather to present comparative data, taken from practice, and submit figures on the strength of these shafts with relation to torsional resistance only. As a section taken through the central portion of the shaft is usually of the most interest to the engineer, on account of the combined twisting and whipping tendency, an analysis

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of these middle sections only is submitted. Table XI gives the chemical analyses of the steels and their S. A. E. equivalents. Table XII gives the length, diameter of center section, polar moment of inertia, maximum fiber stress, and factor of safety in each case. The last two values were obtained by calculations similar to those used in studying the clutch and transmission shafts.

Some of these shafts have a solid, while the others have a tubular section. There does not seem to be any standard practice in regard to this point. In some of the designs examined, the propeller-shafts run inside the torque tubes, which might possibly prevent excessive whipping and buckling, while the other shafts run in the open with no protection. In analyzing the figures, it should also be noted that the

TABLE XI—COMPOSITION (IN PER CENT) OF PROPELLER-SHAFT STEELS

Car No.	S. A. E. Steel No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium
1	1045	0.431	0.590	0.017	0.027	0	0	0
3	3130	0.294	0.540	0.023	0.016	1.20	0.59	0
5	1035	0.405	0.360	0.022	0.031	0	0	0
6	6130	0.270	0.700	0.020	0.035	0	0.90	0.18
7	1035	0.352	0.530	0.008	0.036	0	0	0
8	1045	0.430	0.630	0.015	0.040	0	0	0
13	1035	0.350	0.500	0.015	0.030	0	0	0
14	1035	0.350	0.500	0.015	0.030	0	0	0
2	3130	0.250	0.400	0.020	0.025	1.28	0.56	0
4	1035	0.380	0.510	0.015	0.027	0	0	0
9	1035	0.350	0.500	0.015	0.030	0	0	0
10	1035	0.350	0.500	0.015	0.030	0	0	0
11	1035	0.350	0.500	0.015	0.030	0	0	0
12	1035	0.350	0.500	0.015	0.030	0	0	0

propeller-shafts in the first eight cars listed in Table XII are located to the rear of the transmission and take the maximum geared-up engine torque caused by the lowest transmission-gear reduction, whereas the others withstand the stress of direct engine torque only. The average of the factors of safety in the former case (3.6) is much lower than that (12.3) for the latter.

REAR AXLE DRIVE-SHAFTS

It has been the author's observation that failures in rear-axle drive-shafts are unusually common. It has therefore been interesting in this study to examine the different cars with special attention directed to these shafts to secure their chemical analyses, physical properties and to compute their factors of safety.

In arriving at the figures, it was assumed that the differentials in each rear axle worked sufficiently well to divide the geared-up engine torque equally between the right- and left-hand axles. Tor-

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sional resistance only was considered in the computations, although in several cars, where the semi-floating construction was found, the axles are subjected to bending as well as torsional stresses.

In these cases, however, the axle shafts were found to be specially designed with enlarged portions directly under the main outer bearing, gradually tapering toward the differential, to withstand the bending stresses. This form of shaft usually has its weakest section directly adjacent the differential or in the splined portion that engages the differential gears. These inner ends of the axles appeared in all cases to be proportioned to take the torsional load only. Therefore, sections at this point, in the case of semi-floating axles, were measured and used in securing the data. It was found that axles

TABLE XII—PROPELLER-SHAFT DIMENSIONS AND STRESSES

Car No.	Lowest Gear Reduction	Max. Torque Transmitted Lb. In.	Length, In.	Least Dia., In.		Polar Moment Inertia, Center Section	Max. Fiber Stress, Lb. per Sq. In.	S.A.E Steel No.	Expected Elastic Limit, Lb. per Sq. In.	Factor of Safety
				O.D.	I.D.					
1	5:1	7750	28 23/32	1 1/4	..	0.240	20,200	1045	55,000	2.7
3	3.99:1	5375	54 5/16	1	..	0.098	27,500	3130	65,000	2.4
5	4.3:1	6500	.....	1 1/4	..	0.240	16,900	1035	55,000	3.3
6	4.66:1	6520	57 15/16	1 1/16	..	0.124	27,800	6130	65,000	2.3
7	3 9:1	4625	47 5/8	1 1/4	1	0.141	20,600	1035	55,000	2.7
8	3.63:1	3970	50 3/4	1 5/32	..	0.174	13,200	1045	55,000	4.2
13	3.42:1	9400	44 3/4	1 3/8	..	0.354	18,250	1035	55,000	3.0
14	3.3:1	7400	81 3/4	1 3/4	..	0.920	7,030	1035	55,000	7.8
2	1:1	1450	.....	1 1/8	..	0.157	5,200	3130	65,000	12.5
4	1:1	1010	51 1/8	1	..	0.098	5,155	1035	55,000	10.7
9	1:1	2800	44 1/8	1 9/16	1	0.491	4,450	1035	55,000	12.4
10	1:1	3360	49 7/16	1 35/64	1	0.462	5,630	1035	55,000	9.8
11	1:1	2240	54 5/8	1 9/16	1	0.491	3,560	1035	55,000	15.5
12	1:1	2700	52 5/16	1 9/16	1	0.491	4,300	1035	55,000	12.8
									Average.	....7.3

of the three-quarter- and full-floating constructions were generally made of uniform section between the splined or squared ends engaging the differential gear on the inside and the wheel hub on the outside. In securing the data on these, the smallest diameters were chosen and used in the calculations.

In considering the torsional loads on these axles, it should not be forgotten that the twisting effect of the engine is multiplied many times by the low-speed gears in the transmission and again by the reduction in the driving bevel gears. Take for instance Car No. 1, which has an engine torque of 1550 lb. in. It has a reduction of 5 to 1 in the low ratio gears of the transmission and a reduction of 4 to 1 in the rear-axle bevel gears, or a total of 20 to 1. This total reduction causes a final torque in the axle drive-shafts of 31,000

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lb. in., with one-half of this, or 15,500 lb. in. on each section. In the engineer's effort to keep the total rear-axle weights down to a minimum, the smallest possible shafts are used. High-grade alloy steels must therefore be employed to meet the high-duty stresses imposed by the large gear reductions, but even this practice fails to show up a healthy average for factors of safety.

It can be said that great inconsistency prevails in engineering practice when one compares clutch-shaft to rear-axle sizes. Consider, for instance, the clutch-shaft in Car No. 7. Its smallest diameter is 1⅞ in. to transmit 1160 lb. in. With a total gear reduction of 16.85 to 1, the twisting moment on each section of the rear-axle

TABLE XIII—COMPOSITION (IN PER CENT) OF REAR-AXLE SHAFT STEELS

Car No.	S. A. E. Steel No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium
1	3135	0.359	0.540	0.018	0.038	1.56	0.81	0
2	3125	0.240	0.360	0.012	0.020	1.56	0.52	0
3	3125	0.259	0.460	0.009	0.015	1.11	0.56	0
4	2330	0.310	0.720	0.020	0.020	3.52	0	0
5	3125	0.228	0.680	0.019	0.023	1.10	0.84	0
6	6125	0.270	0.700	0.020	0.035	0	0.90	0.18
7	1035	0.353	0.530	0.008	0.036	0	0	0
8	3135	0.350	0.360	0.008	0.021	1.10	0.52	0
9	Special	0.430	0.550	0.010	0.030	4.50	0	0
10	Special	0.430	0.550	0.010	0.030	4.50	0	0
11	Special	0.430	0.550	0.010	0.030	4.50	0	0
12	3140	0.400	0.600	0.010	0.030	1.20	0.55	0
13	3140	0.400	0.600	0.010	0.030	1.20	0.55	0
14	Special	0.530	0.600	0.016	0.022	1.47	1.11	0

shafts is 9800 lb. in. If the same steel were used in the rear-axle members that was used in the clutch-shaft in this case, and the same factor of safety desired, the axle would have to be of 3.55 in. diameter. This axle would be prohibitive not only on account of its own weight and expense, but also on account of the increased weight and cost of the other parts entering into the complete rear-axle assembly. These conditions have driven the designer, even with the finest of steels, to the use of a low factor of safety. The average for all the cars is 2.6, but several are in the neighborhood of 1.5.

While considering the stresses in rear-axle shafts, the author wishes to point out the fact that the transmission service brake is now practically obsolete, but in the few cases where it is used, the twisting moments on the axle shafts due to the braking loads are greater than those due to the engine torque. It is customary in careful engineering practice to design, in the case of a transmission brake, to meet the maximum twisting moment caused by the brakes locking the rear wheels and sliding the tires on the road surface.

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Car No. 14 has a transmission brake and thus illustrates the possible difference between engine torque and braking torque on the rear-axle shafts. We find from the table that the maximum gear reduction of 32.9 to 1 produces a twisting moment of 36,750 lb. in., with a factor of safety of 4.0 in each axle section. In computing for the values that obtain due to maximum braking conditions, it is necessary to know the greatest load on the rear wheels, the coefficient

TABLE XIV—REAR-AXLE SHAFT STRESSES AND DIMENSIONS

Car No.	Length of Shaft, In.	Least Dia., In. (2×c)	Lowest Gear Reduction	Max. Torque Transmitted, Lb. In.	Polar Moment Inertia, Weakest Section	Max. Fiber Stress, Lb. per Sq. In.	S.A.E. Steel No.	Expected Elastic Limit, Lb. per Sq. In.	Factor of Safety
1	31 9/16	1 7/16	20:1	15,500	0.420	26,500	3135	65,000	2.5
2	....	1 1/8	18:1	13,050	0.157	46,600	3125	60,000	1.3
3	31 1/8	1 1/16	14.5:1	9,750	0.124	41,800	3125	60,000	1.4
4	30 3/8	1 1/16	14.5:1	7,325	0.124	31,400	2330	75,000	2.4
5	31	1 3/8	16.35:1	12,350	0.352	24,200	3125	60,000	2.5
6	31 7/32	1 1/16	16.87:1	11,800	0.124	50,500	6125	80,000	1.6
7	32 9/32	1 1/16	16.85:1	9,800	0.124	42,000	1035	55,000	1.3
8	31 1/2	1 1/16	14.8:1	8,100	0.124	34,700	3135	65,000	1.9
9	30 3/32	1.574	14.65:1	20,500	0.605	26,700	Special	88,000	3.3
10	30 15/32	1.574	11.5:1	19,300	0.605	25,100	Special	88,000	3.5
11	26 1/8	1.574	12.5:1	14,000	0.605	18,250	Special	88,000	4.8
12	30 3/32	1.574	11.7:1	15,650	0.605	20,400	3140	70,000	3.4
13	29 3/4	1.574	14.85:1	20,450	0.605	26,500	3140	70,000	2.6
14	39 15/16	1 15/16	32.9 :1	36,750	1.380	25,800	Special	104,000	4.0
								Average.	....2.6

of friction between the tire and pavement and the diameter of the rear wheels. In the case of Car No. 14, which was a 3-ton worm-driven truck, the greatest allowable load on each rear wheel is 5025 lb. and the diameter of the wheels is 36 in. A coefficient of friction of 0.6 is commonly used for rubber tires sliding on pavement, but owing to the extended use of block tires, and also the common use of anti-skid chains on the rear wheels, a coefficient of friction of 1.0 is a safer assumption.

With a load of 5025 lb. on the tires of each rear wheel and a wheel radius of 18 in. as a lever arm, a twisting moment of 5025 × 18 × 1.00, or 90,500 lb. in., results in each axle section when the wheels are locked by the transmission brake. Compare this with the geared-up engine torque of 36,750 lb. in. and we see that the twisting moment from the brake is 2.45 times greater and the factor of safety is reduced to 1.6. In order to secure even this margin of safety with the shaft diameter used, an exceptionally fine alloy steel with an elastic limit of 104,000 lb. per sq. in. was employed.

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AUTHOR'S CONCLUSION

Table XV was prepared to sum up the data compiled in this paper and to show at a glance all the factors of safety obtained. While the table exhibits several extreme values, the author considers the averages represent values that could be safely adopted as standards for these particular elements in all future engineering practice.

TABLE XV—SUMMARY OF FACTORS OF SAFETY

Car No.	Front Axle	Front-Axle Spindle	Clutch Shaft	Transmission Shaft	Propeller Shaft	Rear-Axle Shaft
1	5.9	20.9	5.8	4.2	2.7	2.5
2	7.3	21.0	81.0	5.0	12.5	1.3
3	4.6	22.2	18.5	4.5	2.4	1.4
4	4.0	9.3	19.0	8.4	10.7	2.4
5	4.1	16.2	23.6	5.5	3.3	2.5
6	8.6	22.9	9.4	4.0	2.3	1.6
7	4.1	26.1	139.5	4.4	2.7	1.3
8	3.9	14.3	10.8	4.2	4.2	1.9
9	5.9	22.3	12.1	13.5	12.4	3.3
10	6.8	40.4	9.8	11.2	9.8	3.5
11	6.9	23.5	14.6	16.8	15.5	4.8
12	5.9	39.6	28.6	14.0	12.8	3.4
13	6.2	41.6	16.3	5.1	3.0	2.6
14	6.9	44.8	14.6	16.8	7.8	4.0
Average	5.8	26.1	28.8	8.4	7.3	2.6

In studying Table XV, attention is directed to the interesting fact that the factors of safety decrease step by step from the clutch-shaft down through the transmission line to the rear-axle drive-shafts. The listed values of all the members affected by the gear reductions are actually lower than would exist in practice because they are estimated on the basis of 100 per cent efficiency for transmission gearing and bearings. A condition of 100 per cent efficiency in any mechanical device is impossible, but the uncertainty of the values of friction and other losses made this assumption imperative. The author believes the theoretical figures for geared-up torque are safe to follow even though they are a trifle high, because the variation in the strength of the parts due to improper heat-treatment could easily exceed the loss due to friction of gearing and bearings.

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CAR PERFORMANCE

By D. L. Gallup

(Member of the Society)

Abstract

The author points out the diversity of opinion on what constitutes desirable car performance in the minds of engineers and of the public generally. He believes this is largely due to the great diversity of claims which have been made in advertising literature and decries the sort of tests which have been made the basis of this publicity, pointing out that a majority of them are conducted under such conditions as make it practically impossible for the car owner ever to duplicate or confirm them.

The kind of an expression or test which will inform the buying public most is one which will tell what the car will do in the hands of the average owner, and define the conditions under which a demonstration of this ability can be made, such conditions to be relatively simple and easy of fulfillment.

In all probability, no subject relating to any part of the field of automobile engineering has caused more discussion, created more dissension, possibly separated more friends or brought about more trouble generally than that which is involved in the title of this paper. There seem to be almost as many interpretations of what constitutes desirable performance as there are individuals involved. This fact, to say the least, is sufficient to indicate a condition of chaos.

In these days of rapid advance in engineering and of accompanying close competition, with all that this implies, it is not surprising that conditions are such that the question "What is car performance?" goes without an answer; this because so much latitude and so many privileges are accorded the advertiser. In a sense both the manufacturer and buyer, particularly the latter, are responsible for this condition. Until the public demands recognition of the fact that its intelligence is equal to that of the advertising manager and that the latter shall not make "wild-cat" claims of behavior for the product of his company, there will be no diminution of the perplexity a man now goes through in endeavoring, previous to purchase, to make logical comparisons between one car and another.

The advertising during the last few months has with many people put the finishing touches on the situation and although it is hardly to be expected that much of a corrective nature can be accomplished, this paper is a mild protest, made in a formal way, against the existence of fifty-seven varieties of schemes for entrapping the buyer.

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TESTS CANNOT BE DUPLICATED

So many "tests" have been made, so many statements as a result of these "tests" have been given publicity, and each is in so many ways so far removed from the possibility of duplication by the buyer, that he at times doubts if there can be any relation between what he owns and what he has been told he owns. There is no excuse for this. The American buying public is composed for the most part of men who know their business thoroughly, and who can be trusted with the truth, which to them is always preferable to a contention impossible of demonstration. There is no reason, positively, why a man when he buys a car should not expect to be able to duplicate in every detail whatever performance claims have been made by the agent or manufacturer.

Nothing in modern business should require a man to seek the services of a specialist to treat his car so that such performances are possible, the consequence being to render the machine unfit for natural purposes. If a device cannot find a market without being heralded by advertisements that have for their purpose the blinding of the reader to possible detriments or to the good points in other manufacturers' product, then it can be said truly that the device has no reason for being sold, and its manufacturer had better seek other fields. This may to some seem somewhat radical, but when it is realized that the substantial builder is he whose product speaks for itself and that as has so often been said, "The recollection of quality remains long after the price has been forgotten," one can see that it will be but a short time until the public will demand advertising permitting of but one understanding, and that the correct one.

All of the foregoing embodies merely the reflections of one who has attempted to analyze the various feats given publicity in the press and credited as being representative of standard products. It is also a long and much involved way of saying that, in most instances, these so-called "stock car" performances are no more stock-car performances than that a single and a twelve-cylinder engine are the same.

From the little curved-dash runabout with a single cylinder of 5 or 6 hp., which worked just often enough to cause us in our enthusiasm to overlook its faults, to the modern car with a multicylinder engine of upward of 90 to 100 hp., so smooth in action and general behavior and so capable in performance that one can hardly believe the two cars represent the same general idea, is a long way in point of performance, although not so long as far as time is concerned.

TREND OF DESIGN

An actual comparison of car capacities at both engine and rear wheel has been available as a result of the equipment at the Worcester Polytechnic Institute, and shows a steady trend in a single direction and one apparently universally accepted without question by all of the

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representative automobile manufacturers. This trend, which is the increase in horsepower output with a reduction in weight and fuel consumption, is evidenced to such an extent that a common focus seems about to be reached. In other words, in the various classes of vehicles, the ratio of delivered horsepower to weight of car is approaching a common basis. That this has not always been the case is well represented by a paper,* read before the Society in 1914 by the author, which emphasized the wide diversity of results obtained at that time. Nowadays, however, *all* things considered, car performance is nearer a common standard than ever before, because competition has forced engineering and sales to a point of at least "near" perfection.

The statement in the preceding paragraph may seem somewhat inconsistent in view of the general idea of protest embodied in this particular discussion, but appears so simply because at present variation in performance in many instances is due to cleverly manipulated accessories or details, resulting in claims misinterpreted by the public, and which from a standpoint of *moral* legality have no place in a comparison.

The mere fact that a certain make of engine, when placed in a given chassis fitted with special wheels, tires, gear ratio, lubricant, fuel, valve-timing, carbureter, ignition system or other accessories, will do 100 m.p.h. or more for so many hours upon a specially constructed speedway is *not* of particular interest to the buyer of the car fitted with that same general form of engine. Such special conditions do not exist in what he buys and what is more are not capable of duplication or trial by him in the pursuit of ordinary driving. True, he may boast that "his" car *can* do these things, but he knows in his heart that the only facts admitting of discussion are the dimensions of bore and stroke.

What the average man is interested in, previous to purchase, is a true statement of behavior, one that he can expect to realize in driving his car, and, what is more important, one that he can use as a basis of comparison. He then finds it necessary to take into consideration price, passenger capacity, quality of equipment, appearance, use to which the car is to be put, length of time to be possessed before discarding, and other things too numerous to mention, the relative importance of which will vary with nearly every individual.

Inflated values as to speed, acceleration, economy (relating to fuel and lubricant) and hill-climbing ability can be obtained separately by the manufacturer with the special equipment required to bring out the respective points. In reality such values do not apply to his cars as a *class* and have no place in real engineering and should be guarded against carefully. What *is* desired, is, as has already been sug-

*Automobile Effectiveness, Prof. D. L. Gallup. S. A. E. TRANSACTIONS, Part I, 1914, page 165.

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gested, a true statement of ability, one which applies to *any* car and which the owner can count on obtaining himself if he so desires.

REASONABLE ADVERTISING

Probably a time will never come when manufacturers will agree that reasonableness in advertising practised consistently is of more genuine value than extraordinary claims of "stunts." On the other hand, there is no question but that in the near future an official request will be made for some form of test for "car performance" that will take into account all of the variables affecting such performance, and that can be conveniently used.

This paper then serves a double purpose: One is that of a warning and the other that of a basis for discussion as to what items properly belong in such a formula as may be suggested and eventually adopted.

Why is the S. A. E., through one of the Divisions of the Standards Committee, now endeavoring to establish a form of test for gasoline consumption and also one for acceleration ability?

Simply to obviate "freak" tests with their attendant misinformation, and also to furnish the conservative engineer with information of value to him in rating his product. Even more necessary is a formula for "car performance"; one that will enable the engineer or even the prospective purchaser to "rate" a given car and thereby establish a means of comparison; one dealing particularly with the overall efficiency or, better yet, with the dollars and cents for upkeep.

This should take a form similar to what has been prepared a number of times by those interested in this same subject, except that in the opinion of the author it should go still further. Such a formula should take into consideration all of the items bearing directly upon the features involved in performance. It should be representative of actual, not theoretical data. To be more specific, should it be desirable to involve in the formula something relating to capacity, it should be in the form of displacement rather than of horsepower: the former is a function of efficiency; the latter when regarded alone is significant of nothing in particular.

It would be unwise at this stage to suggest other than tentatively the nature of the final formula. Merely may it be said what it should at least contain. Previous to doing so, however, let a suggestion be made as to the details of a test.

DETAILS OF TEST SUGGESTED

In the first place, the test should be made with a stock car, and this regularly equipped. In the author's opinion a "stock" car is one such as might be picked at random in *any* salesroom in the country, not one fitted with special gear ratio or the other details already mentioned and certainly not one that has been specially tuned for test.

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Again, all dimensions should be recorded. These would include information regarding the bore and stroke, valve-timing, gear ratio, tires and weight, in fact, everything in any way affecting the results, and of value in identifying the car as "stock."

The test itself might be of three kinds: One, on the level and at definite speeds; another, on a grade; and a third, an acceleration test on the level. The first and third are already fairly well taken care of by suggestions now under consideration by the Research Division. The second could be easily worked out. From these tests curves or values could be deduced representing a performance capable of comparison with that of another car. Such items, of course, as involve the individual pleasure of the owner should not be considered.

Coming now to the particular items, the following should be included: (1) Weight of car and passengers; (2) displacement of engine; (3) speed in miles per hour; (4) miles per gallon of gasoline; (5) gear ratio; (6) size of tires; (7) maximum brake-horsepower of engine; and (8) grade in per cent.

The "certificate of performance" would include a complete record of gasoline consumption as accomplished under S. A. E. specifications, the same for the acceleration test, and a curve or series of factors for various speeds as determined by the formula and having reference to tests on the level or on a grade.

For the present this is as far as it is dared to go. It is hoped that there will be enough in the form of suggestions or criticisms on which to base a substantial argument for a definite formula and if this is done it will be felt that the mission of this paper has been accomplished. In addition, and of much more importance, it is believed that much of the uncertainty now due to misrepresentation will disappear and the industry will be given a considerable impetus for the simple reason that competition will become more keen and the ultimate aim will be reached just so much sooner.

DISCUSSION

JOHN YOUNGER:—A formula that takes into account only the physical dimensions of certain parts of a car, with the idea of rating its ability, is unreliable and unfair. It is unfair to the manufacturer who has spent much thought on the development of a more efficient performance from the dimensions given. It is unreliable, as a poor out-of-date car will be rated the same as the latest model, provided both have the same dimensions. A formula should always indicate a car's actual ability, and the problem arises at once as to what can best express this ability.

Hill-climbs are often unreliable, as road and wind resistances vary, and grades are usually overestimated. Gasoline consumption in miles per gallon is probably the most easily measured actual performance and has the added advantage of giving a measure of economy.

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If gasoline consumption is measured, not as in freak tests, but with the ordinary driver over several hundred miles of ordinary roads, a fair idea is obtained of what a car's economy is in this respect. Coupling this with the dimensions of engine, the gear ratio and the wheel diameter, a reliable formula for rating a car is obtained.

H. Kerr Thomas' Sigma formula, as proposed in *Internal Combustion Engineering*, April 2, 1913, includes all these factors:

$$\Sigma = \frac{0.0047\, d^2\, s\, N\, g\, M}{D}$$

where d = engine cylinder diameter, in; s = length of stroke, in; N = number of cylinders; g = total gear ratio; M = gasoline consumption, miles per gallon; and D = driving-wheel diameter, in.

This formula shows that if a car is made with large cylinder capacity and low gear ratio to enable it to work under heavy physical conditions, the gasoline consumption will be heavy in long-distance work. If, on the contrary, gasoline consumption is considered everything, to the exclusion of ample power for ordinary road conditions, the decrease of $(d^2\, s\, N\, g \div D)$ automatically reduces the value of Sigma.

The formula places a premium on the manufacturer who from a given $(d^2\, s\, N\, g \div D)$ can get superior gasoline economy, and it is only right that this should be encouraged.

I would, therefore, suggest strongly that this formula be considered for rating all vehicles.

A few Sigma values are:

Rolls-Royce, 50 hp., 1911, R. A. C. trial	4.70
Napier, 60 hp., 1912, R. A. C. trial, 1912	4.27
Rolls-Royce, 50 hp., on average roads around Buffalo	2.60
Maudslay 4-ton truck, 1907, R. A. C. trials	2.70
Pierce-Arrow 5-ton truck in Buffalo service	2.60
Pierce-Arrow 38 in 1912	2.28
Pierce-Arrow 48 in 1912	2.24
Pierce-Arrow 66 in 1912	2.16
Pierce-Arrow 38 on several hundred miles hilly country, 1912	2.00
Sunbeam 30 on same route at same time	1.69
Pierce-Arrow 66, average of a long tour in California	2.62
Sunbeam, 30 hp., 200 miles run, Buffalo-Toronto and back, 1913	2.20
Pierce-Arrow 2-ton truck in two weeks' N. Y. service	2.57
Pierce-Arrow 5-ton truck on 100-mile run, good road	2.91
Ford, 1913 model	3.84
Thomas-Leyland-Delahaye electric transmission truck, R. A. C., 1911	3.56

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POSSIBILITIES OF THE CONSTANT PRESSURE CYCLE

By Arthur B. Browne and Herbert Chase

(Members of the Society)

Abstract

The authors first define the elementary conditions governing combustion efficiency, dividing these conditions into three main classes. They next compare engines operating on constant volume, constant temperature and constant pressure cycles, dealing specifically with the Otto, Diesel and semi-Diesel types.

The main part of the paper is devoted to an outline of the constant pressure cycle, analyzing its advantages as compared with the merits of the constant volume cycle now used in internal-combustion engines. The paper is concluded with a detail description of a proposed constant pressure engine.

LIMITATIONS OF OTTO CYCLE

Have not engines designed to operate on the Otto or constant volume cycle reached a state of development such that but little further progress of moment is to be expected in the art of their construction, save perhaps in refinement of mechanical detail? Or, to put the question in another form, are not the limitations of the Otto cycle as now applied, notably its poor thermal efficiency under the average low compression pressures realized in practice, such as to force the automobile engineer to consider carefully the possibilities of other cycles, if the art of motor-car construction is to progress as rapidly in the future as in the past?

These are fair questions and must in the opinion of the authors be answered in the affirmative. Having so concluded we have given much thought and study to the subject of this paper and have found it to embrace possibilities which we believe are certain to have far-reaching effects in the entire field of the engineering of internal combustion prime movers.

Before considering the possibilities of the constant pressure cycle it will be well to consider, for purposes of contrast and comparison, the advantages and disadvantages of engines operating on other cycles.

The thermal efficiency of any cycle is dependent in large part upon the conditions under which combustion takes place. Since these conditions vary greatly as between cycles it is most important to have clearly in mind, when making a comparison, the factors conducive to maximum combustion efficiency.

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COMBUSTION EFFICIENCY AND CLASSIFICATION

The liberation of heat energy in a gas engine depends for its efficiency upon several conditions that can exist prior to ignition and during the combustion period. The efficiency is a maximum when the following conditions obtain:

(a) The density or compression of the charge is the greatest possible.

(b) The temperature prior to ignition is the highest possible.

(c) Oxygen is present in quantities just sufficient for complete combustion.

(d) The admixture of fuel with the necessary oxygen is perfect, that is, when the charge is absolutely homogeneous.

(e) Inert and diluting gases are absent.

Combustion can be divided into three main classes.

In *Class A* all the oxygen is furnished by the supporting atmosphere without admixture prior to combustion. An illustration of this is the luminous gas flame, in which gas issuing from an orifice finally comes in contact with the oxygen of the air and burns with a luminous flame of low thermal efficiency. It will be noted that Class A fulfills none of the conditions previously enumerated as making for efficiency because: (a) Both gas and air are at atmospheric pressure; (b) The temperature prior to ignition is low; (c) Much more oxygen is present adjacent to the combustion zone than is necessary and this excess is heated without useful purpose; and (d) There is no intimacy of the mixture, the gas molecules being forced to "seek" their proportions of oxygen before combustion can occur. Hence in the luminous gas burner the gas is caused to issue in a thin sheet, whereby its surface is made as great as possible in proportion to its volume.

In *Class B* part of the oxygen is mixed with the fuel, the remainder being furnished by the supporting atmosphere. This class is illustrated by the Bunsen burner and its thermal efficiency is, obviously, far greater than that of Class A. In this case the first three conditions are not complied with, the increased efficiency being clearly due to a partial compliance with condition (d).

The change in efficiency obtainable as the other conditions are complied with is illustrated by the blowpipe, where the pressure on or density of the charge is increased and a marked increase in efficiency of combustion results, while with the superheating air coil sometimes used in connection with the blowpipe, the use of which tends to comply with condition (b), still higher efficiencies are obtained. It is probable, however, that the combustion in the case of the blowpipe can be considered in some measure as falling under Class C.

In *Class C* fuel and air are mixed in such proportions that there is just sufficient and no excess of oxygen, so that combustion ensues without regard to the gaseous medium in which it takes place. This is sometimes called "flameless" combustion, from the fact that the

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combustion is so rapid and so thorough that it takes place by concussive propagation between the molecules, in a very limited zone, almost a sheet, which is termed the "flame cap."

The term "surface combustion" is sometimes used interchangeably with "flameless combustion." In the opinion of the authors all combustion under Classes A and B is "surface" combustion; that is, the reactions take place only on the *surface* of the gas, when it meets the necessary oxygen.

Class C, however, embraces combustion within the mass—a true molecular interchange confined to the sheet of flame cap only because at that point alone the rate of propagation between the molecules exactly balances the rate of flow.

In the case of flameless combustion there is no dilution of the unburned gases by the burned. The mixture of gas and air approaches the flame cap, complete and instantaneous combustion takes place and the products of combustion move away. Thus the surface on which combustion takes place is the dividing point between burned and unburned gas. The process is orderly and efficient to the highest degree.

If a mixture of combustible gas and air in correct proportions is subjected to compression and heated prior to ignition the first four conditions (a, b, c and d) are present and combustion of the highest efficiency obtainable in the presence of an inert gas, such as nitrogen, ensues.

COMPARISON OF ENGINE CYCLES

Internal combustion engines can be classified as follows:

Class I.—Engines operating on constant volume cycle.
Class II.—Engines operating on constant temperature cycle.
Class III.—Engines operating on constant pressure cycle.

The Otto Cycle

To the first class belongs the Otto cycle, applied universally at the present time for internal-combustion automobile engines. Its chief advantages are its flexibility and adaptability to relatively light-weight and therefore to easily portable units. Its disadvantages are many and include the following:

(1) Poor thermal efficiency under the average condition of low compression pressure, which results from throttling of the charge and which (at maximum) must be relatively low to prevent self-ignition when a fuel rich in hydrogen (such as gasoline) is used.

(2) High explosion pressures occurring so suddenly as to deliver what practically amounts to a hammer-blow on the piston head. To meet this condition the parts must be much stronger (and heavier) than they would otherwise need be to accommodate the relatively low mean effective pressure produced.

(3) Large clearance space required to admit of the low compres-

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sion pressures necessary with rich and volatile petroleum fuels, which are practically the only fuels commercially available that will give reliable operation under varying load conditions. This clearance space is always filled with burnt gases, which dilute and expand the incoming charge of unburnt gas.

(4) Impracticability (especially in light high-speed units) in cases where only heavy or relatively non-volatile fuels are available. This applies only to oil engines; that is, not to gas engines.

Combustion in the Otto cycle is superior by reason of its partial compliance with conditions (a, b, c and d) above. Combustion in this cycle clearly falls under Class C and would be highly efficient were it not for unavoidable losses and inherent limitations. But under condition (a) we find the density of the charge is limited by liability to pre-ignition, and under (b) the same limitation is placed on the pre-ignition temperature; as to (c) and (d) it is to be noted that the proportions and intimate admixture of the fuel and air depend on the efficiency of the carbureting device used. This is never perfect and is usually far from ideal. From condition (e) it is evident that the presence of burned gases in the charge not only serves to decrease the unit weight of charge taken in but entails a direct loss because of the heat delivered to these products of combustion.

The Constant Temperature Cycle

No engines practical for motor vehicles have ever been developed using this cycle (Class II). Engines operating on the Diesel cycle were originally intended to operate on a constant temperature cycle—in fact, the Diesel patents so stipulated. In practice, however, it has been found that Diesel engines operate more nearly on the constant pressure than on the constant temperature cycle.

The Diesel Cycle

The Diesel cycle has several inherent advantages. High thermal efficiency, adaptability to use of heavy oils, variable cut-off (within certain limits) and high compression temperatures, making ignition devices unnecessary, are foremost among these. Its disadvantages, on the other hand, are quite as numerous and in some cases are insurmountable where small light-weight units are required. These disadvantages have to do largely with the high pressures encountered and include the following: (1) High compression and explosion pressures making necessary heavy and close-fitting parts, difficult to keep tight (engine can become inoperative if not tight); (2) Necessity for high pressure fuel injection pump, and separate air compressor; (3) Mechanical difficulties in regulating the minute quantities of fuel discharged by the pump to accommodate different loads; (4) Fuel admission line limited by the relatively small volume

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(entailed by the high compression necessary) to be heated by combustion; (5) Difficulties in starting with resultant complication; and (6) Low mechanical efficiency owing to close-fitting parts (especially piston rings.)

The combustion conditions applying in the Diesel cycle, as exemplified in modern practice, approximate closely those of Class B. The latest practice in this cycle is to atomize and inject fuel by highly compressed air; hence combustion is dependent only in part upon the compressed air within the cylinder. The high combustion efficiency of this cycle is, however, due largely to compliance with condition (a) and to the partial fulfillment of condition (d).

The Semi-Diesel Cycle

The hot-bulb or so-called semi-Diesel engines possess most of the advantages of the Diesel type, although they are less efficient because of the lower compression pressure employed. They are limited as to power output per unit of weight on account of disadvantages similar to those of the Diesel cycle. This alone would preclude their adoption for motor vehicles.

As compared to the Diesel type the decrease in thermal efficiency and the fact that some form of ignition device must be provided are offset by lower compression pressure and the attendant advantages of somewhat lighter parts, and less expensive machine work. The thermal efficiency of the semi-Diesel engines is higher than that of the Otto cycle engines, but this efficiency is obtained at the expense of heavier parts made necessary in part by the high pressures encountered. Largely because of this fact and of difficulties and complications incident to the problem of starting (such as preheating of the bulb), the semi-Diesel type of engine has never as yet been applied commercially in motor-vehicle construction.

Combustion in the semi-Diesel cycle falls under Class A. Fuel is injected into compressed air and each molecule must separately seek its individual quota of oxygen. Intimate admixture, condition (c), is impossible, and carbon deposits are the result. But since the temperature and density of the charge are high, the thermal efficiency is better than that of the Otto cycle.

The Constant Pressure Cycle

Let us now consider engines belonging to Class III. The advantages of this type are numerous and the disadvantages encountered in the past have had to do almost entirely with constructional difficulties. While the former have long been appreciated, not to say regarded as ideal, the latter have stood in the way of progress and have operated against the development of any commercially prac-

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tical* engine applying the principle of this cycle—this excluding, of course, engines operating on the Diesel or semi-Diesel cycles as not, strictly speaking, belonging to the constant pressure cycle class (see previous reference to Diesel cycle engines, under heading "Constant Temperature Cycle."

In order to have in mind exactly what is meant by the constant pressure cycle let us first consider the succession of events. A theoretical card for this cycle is shown in Fig. 1, in which *BC* represents adiabatic compression from atmospheric to maximum pressure;

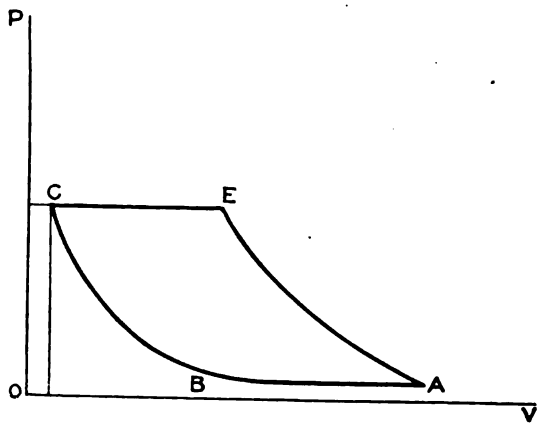

FIG. 1—THEORETICAL CARD FOR CONSTANT PRESSURE CYCLE

CE, addition of heat isopiestically; *EA*, adiabatic expansion from maximum pressure to atmospheric pressure; *AB*, cooling at atmospheric pressure. Both heating and cooling are effected at constant pressure, and the compression pressure is the maximum pressure of the cycle. There is no sudden rise in pressure at ignition as in the Otto cycle.

ADVANTAGES OF THE CONSTANT PRESSURE CYCLE

High Mean Effective Pressure With Low Maximum Pressure

(1) Inspection of the card given in Fig. 1 at once makes evident the two cardinal advantages of the cycle: (a) the large area indicating

*Messrs. Carpenter and Diederichs state in their book on the Internal Combustion Engine that the Brayton oil engine was a thoroughly practical engine and saw wide application. This was true in its day but the engine has long since been abandoned, partly because of its inefficiency as compared to the modern Diesel types, but more especially because of cumbersome design and because its inventor failed to use certain important details of construction that will be treated of later.

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high mean effective pressure and (b) the low maximum pressure. In practice these are factors of utmost importance. They result in high and relatively uniform torque, large power output per unit of displacement and relatively low maximum bearing pressures and unit stresses on parts, with consequent light weight and length of life.

Variable Cut-Off With Constant Compression Pressure

(2) The possibility of varying the point of cut-off is practically the same as that in the ordinary steam engine. The variation can be made through such a wide range that the engine accommodates itself to most if not all variation in load. This fact is of utmost importance, since it operates to make the cycle more efficient at part load than it is at full load. No corresponding variation to meet changes in load is possible in the Otto cycle unless with "hit or miss" governing, which of course is not feasible for automobile engines. Variations in load are taken care of as a rule by throttling of the charge with consequent decrease in compression pressure. The thermal efficiency in any cycle falls rapidly with a decrease in volume ratio. In the automobile engine a maximum thermal efficiency of about 20 per cent is obtainable under full-load conditions; the thermal efficiency falls as low as 2 or 3 per cent under light load when the charge is throttled and the compression pressure correspondingly reduced. Since automobile engines run throttled during a large part of the time their average thermal efficiency is necessarily low. Inspection of the card of the constant pressure cycle, Fig. 1, shows the feasibility of making the admission line *CE* so short (by means of an early cut-off) as to cause the expansion line *EA* to fall practically on the compression line *BC*. By this means it is probable that the lightest loads can be carried at the maximum pressure of the cycle with the same fuel efficiency as or higher than at full load. This being the case, throttling of the charge would take place only in starting.

Adaptability to Use of Low-Grade Liquid Fuel

(3) The importance of this factor in view of the constantly advancing price of highly refined petroleum distillates is readily apparent. The method by which low-grade fuel can be utilized will be outlined later. But engines of this type can be made to run on quite as heavy an oil as can be utilized in any Diesel engine and with far better combustion conditions than are possible in the Diesel type. Self-contained constant pressure engines can therefore be expected to replace those of the constant volume type in the small size light weight class as rapidly as the Diesel engine has replaced the Otto cycle engine in the heavy weight class, where fuel cost is much in favor of the former. It is not at all unlikely that even the highly efficient Diesel engine will give place to the constant pressure type of engine when its advantages over the Diesel type are fully realized.

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Excellent Scavenging Properties

(4) The scavenging in a four-stroke constant volume engine is never complete because of the large clearance space required. This results in several drawbacks, among which is a loss in volumetric efficiency and a dilution of the incoming charge with inert gases. These faults are not present in constant pressure engines, because an excess of air can be forced through the cylinder while the piston is at or near the lower dead-center.

Volumetric Efficiency

(5) In the constant pressure engine using the working cylinder for a compressor, losses from decreased volumetric efficiency can be rendered very small. The air that can be normally pumped will usually be in excess of the requirements of the working stroke on account of the subsequent expansion by heat. Hence there is as a rule an excess of compressed air from which to draw on the working stroke. The piston cannot move until the volume behind it is sufficient to afford the necessary pressure. For peak loads the normal maximum pressure can even be temporarily increased. This is in marked contrast to the constant volume cycle, in which the volumetric efficiency is not high, even at low speeds, and usually decreases as the speed increases.

Adaptability to Operation on the Two-Stroke Cycle

(6) Constant pressure engines will operate much more satisfactorily on the two-stroke cycle than will engines of the constant volume cycle, because scavenging in the former can be more nearly perfect, through the use of an excess of air to remove the products of combustion, whereas unless fuel injection with its inherent difficulties is resorted to, engines of the constant volume two-stroke type must depend upon the incoming charge to scavenge the combustion chamber. This inevitably results in a loss of fuel. Under these circumstances a four-stroke cycle is unnecessary in a constant pressure engine and all the advantages of the two-stroke cycle are realized.

No Fuel Injection Pump Necessary

(7) The operation of engines of the constant pressure cycle is not dependent upon any fuel injection pump as is that of engines of the Diesel and semi-Diesel type. Aside from the purely mechanical difficulties and complications of such pumps, the metering of the fuel for varying loads presents some exceptional difficulties, especially in small units. From all such difficulties the constant pressure engine is free, an advantage worthy of special mention in comparing the strictly constant pressure engine with one of the Diesel or semi-Diesel type.

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No Starter Necessary

(8) In practically all types of Diesel engine some form of high compression air starter is necessary on account of the high compression pressures attained. No such complication is necessary in the constant pressure engine. A small quantity of air under a pressure of perhaps 1 to 5 lb. will be sufficient to start the engine. When once started the engine will quickly pump air up the predetermined maximum of the cycle and the pressure will then remain constant.

Combustion Conditions

(9) The constant pressure cycle comes closest to fulfilling all the conditions for efficient combustion. Combustion takes place under Class C conditions. It is highly efficient because: (a) The density of the charge is limited by structural considerations only; (b) It is possible to raise the temperature of the compressed air to a high degree, prior to the introduction of fuel, by utilizing the exhaust heat (which is commonly wasted); (c) The proper proportions of fuel and air can be automatically maintained without mechanical complications; (d) On account of the appreciable time that elapses between the entrance of the fuel and its final combustion, its complete vaporization and diffusion, even though it be of low grade, is assured by its introduction into the highly heated air, noted in (b) above; and (e) As was seen under heading of "scavenging properties" little if any dilution is due to presence of burnt gas in the mixture so that except for the presence of atmospheric nitrogen, which of course cannot be excluded, dilution is at a minimum.

REASONS FOR SLOW DEVELOPMENT OF CONSTANT PRESSURE ENGINES

Consideration of the numerous advantages of the constant pressure cycle immediately raises the question "why has not the cycle seen a wider practical application?"

The answer to this question can be found in these circumstances: During the period 1872-77 Brayton produced the first commercially successful constant pressure engines. These were built early in the art and were extremely inefficient, as judged by modern standards, both mechanically and thermally. They were used to some extent, but were replaced by engines of the Otto cycle after the year 1877, when Otto took out his American patents. The greater efficiency of this type *as then constructed* reacted against the Brayton engine, which would, if better understood and designed, have readily held its own or displaced the Otto engine. Brayton's engine was cumbersome and so designed as to require a separate compressor. This resulted in low mechanical efficiency and complication, which could have been avoided entirely, as will be seen later.

The advent of the Diesel engine, which proved to be much more efficient thermally than the Otto engine, was another element in

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diverting attention from engines of the strictly constant pressure type. Under present circumstances there appears to be no reason why engines much more efficient than those of the Otto type and lighter than those of the Diesel and semi-Diesel types should not be readily produced. A *purely diagrammatic* representation of such an engine is shown in Fig. 2. This should *not* in any case be considered as indicating *actual* or *recommended construction* except as to *general principles* involved.

PROPOSED CONSTANT PRESSURE ENGINE

The cylinders *1* serve as both the compression and working cylinders in which move the differential pistons *2*. Air first enters the compression spaces *3* of the larger diameter of the pistons through port *4* on the in-stroke and is compressed on the out-stroke. This air need never be raised to more than 5 lb. pressure, and this can be attained by other means than a differential piston if more desirable. The air thus compressed serves to scavenge the adjacent cylinder, entering the latter via port *5*. The products of combustion leave the cylinder through port *6* (unless some other valve be provided in or near the cylinder head as a better means of exit). As the pistons move on the outward stroke, ports *5* and *6* are closed and the air remaining in the cylinders is compressed and discharged through the valves *7* into the receiver *8*.

Since the clearance space between piston and cylinder head is practically zero, substantially all the air is expelled on the out-stroke. Just as the crank passes top-center the admission valve *9* is opened and the mixture is admitted to the burner through piping *10* (where its temperature has been raised by contact with the hot inner piping *11* through which the exhaust gas, discharged after the previous working strokes, has passed). The mixture, still under high pressure, passes into the burners *17*, which are in reality a part of the combustion spaces, as the pistons are forced downward. In the burners (the construction of which will be fully described later) ignition by spark from plugs *19* occurs and *complete combustion at constant pressure* ensues until the mixture is cut off as a result of closing the admission valves *9*.

Action of Exhaust Gases

During the admission period the heat gradually liberated as a result of combustion enables the products of combustion to expand *without loss of pressure* and thus do work on the piston. After cut-off the hot gases expand, with decreasing pressure, and continue to do work until the exhaust port opens. The burned gases, still at high temperature, then pass through the pipes *11* provided for this purpose, and give up to the walls of these pipes and the compressed air surrounding them a large portion of their heat before escaping to the atmosphere. Thus there is saved to the succeeding cycle much

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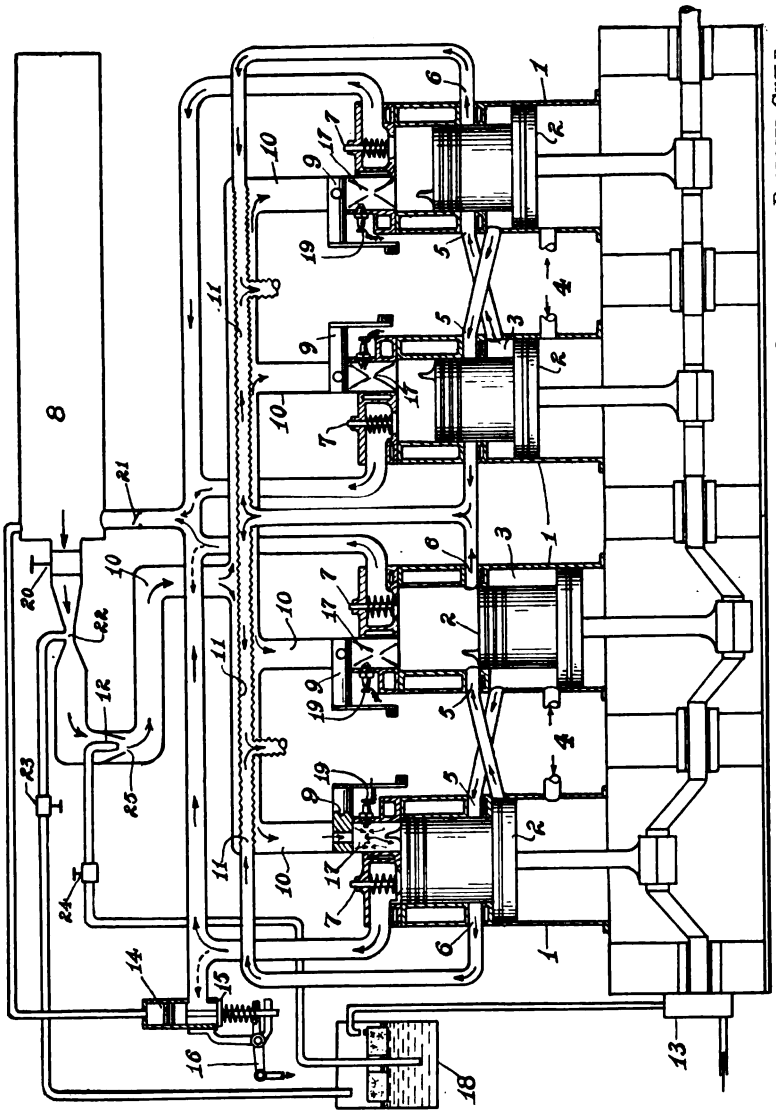

FIG. 2—DIAGRAMMATIC REPRESENTATION OF FOUR-CYLINDER ENGINE OPERATING ON PROPOSED CYCLE

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heat that would otherwise be entirely lost, as it is in most if not all other types of internal combustion engine. Furthermore, the addition of heat to the air in the piping *10* takes place *after* the air has been compressed. Thus its temperature is raised with corresponding increase in ability to do work.

Suppose now the compression pressure decided upon as most desirable be assumed, for the moment, to be 150 lb. Even at normal full load with cut-off at say one-third stroke it is evident that *all* the air compressed in the working cylinder cannot be utilized. Hence the pressure in the receiver will build up rapidly unless some relief valve be provided. To put this on the receiver would, of course, mean the loss of much of the work of compression. Hence a single unloading valve *15*, operated by piston *14*, is provided. When the pressure in the receiver rises above the 150 lb. desired, the valve *15* is forced open against its spring and the air in the cylinder is simply discharged to atmosphere with only slight loss of power.

If for any reason a momentary overload is to be carried by the engine some device such as *16* for increasing the tension on the spring that normally holds the unloading valve closed, can be brought into play. In the case of a motor-car engine such a device could be operated by a simple dash control.

Method of Supplying Fuel

Fuel is supplied to the engine as follows: The pump *13*, positively driven from the engine, draws the fuel from the main supply tank and delivers it to the reservoir *18* in which a float or some other device maintains a constant level. Any surplus fuel pumped is bypassed or returned to the main supply tank. The small tank *18* is maintained at a pressure somewhat less than receiver pressure, depending upon the air velocity through the restricted area *22*. From this tank the fuel is drawn through the spray nozzle *12* by the injector action of the air passing the nozzle. The flow of fuel will of course cease immediately when the air flow ceases on account of the closure of the admission valves *9*. A correct proportioning of fuel to air can be accomplished by proper adjustment of the regulating valves *23* and *24*, Fig. 2. As the velocity through the atomizer *25* increases the natural tendency toward over-richness is counteracted by the proportionately diminished pressure in the restricted area at *22*.

The admission valves *9* can be made of the Corliss, slide or poppet type, as proves most desirable, and be operated by any conventional cut-off device such as, for example, is used in steam engine practice. In the case of a motor-car engine the point of cut-off would be varied by a device operated in precisely the same manner as is the throttle on an ordinary Otto cycle engine. The throttle valve *20* shown in Fig. 2 would be used only in starting.

The check valve *21* prevents air in the receiver escaping to atmosphere when unloading valve *15* is open. The admission valve

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can, if desired, be left open during practically full stroke when a heavy torque is required. The card would then be practically square and resemble closely a card from a steam pump. Under this condition the fuel consumption would of course be much increased because the gases would not be allowed to do work in expanding after cut-off. The periods when such a late cut-off might be used would be short in an engine properly proportioned to the load.

The striking similarity of the constant pressure cycle to that of a steam engine is at once apparent. But while the results are equal in every way to those accomplished with the steam engine, the engine is self-contained and does not require the boiler, condenser and other elaborate external apparatus necessary in the case of the steam engine. An engine operating on the proposed cycle has all the advantages of the steam engine without any of its disadvantages and at the same time possesses characteristics that should render it much more efficient and practical for motor vehicles and many other types of service than is any type of internal combustion engine now in use.

METHOD OF SECURING COMBUSTION

The operation of the proposed cycle is dependent to a large extent upon the functioning of the burner marked *17* in Fig. 2 and shown in detail in Fig. 3. To understand the operation of this burner it is necessary first to have clearly in mind certain fundamental laws governing flame propagation. Imagine a tube composed of material that is a non-conductor of heat, this tube being closed at one end and open to atmosphere at the other. Now suppose the tube be filled with a highly combustible mixture of air and gas. If the mixture be ignited near the open end of the tube the flame will travel toward the closed end at a rate of speed dependent chiefly on the quality, temperature and pressure of the mixture.

Suppose now a vessel containing a combustible mixture under pressure be connected to the open end of the tube. If the end of the tube formerly closed is then opened the combustible gas in the vessel will flow out through the tube at a rate dependent upon the pressure. If now the mixture be ignited at a point midway of the tube the flame will propagate itself either toward the vessel or away from it according to the relation between the velocity of the gas and the rate of flame propagation. If the rate of flame propagation be greater than the velocity of the gas through the tube the flame will travel against the flow of the gas and ultimately enter the vessel from which the mixture is issuing. If the velocity of the gas is greater than the rate of flame propagation the flame will travel with the flow of gas and ultimately blow out at or near the open end of the tube. If, however, the rate of flame propagation is equal to the velocity of the gas the flame cap will remain stationary, the combustible gas approaching it from one side and the products of combustion leaving on the other.

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Description of the Burner

In the case of the burner shown in Fig. 3 the combustible mixture enters under pressure through the pipe *A* and fills the annulus *H* (called the diffusion chamber) surrounding the combustion chamber *B*. Entrance to the latter is afforded by openings *C* so arranged that the streams of gas come from opposite directions and meet at a point where their velocity is zero. The velocity at the point of entrance to the combustion chamber of the burner will depend upon the pressure difference between the combustion chamber and the chamber from which the gas issues. Suppose now the pressure difference is such that the velocity at the point of entrance is 100 ft. a second and that

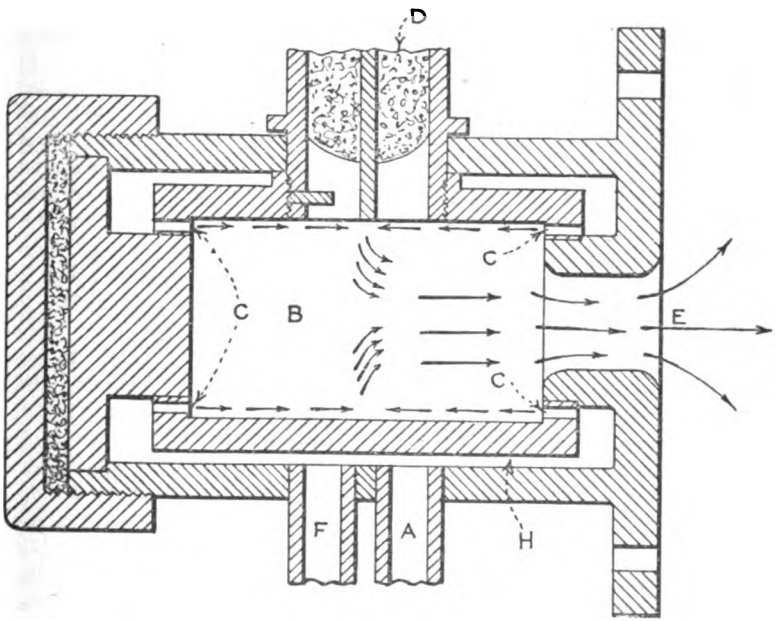

FIG. 3—BURNER FOR PROPOSED CONSTANT PRESSURE ENGINE

the rate of flame propagation in the particular mixture under consideration is 50 ft. a second. If the gas be ignited by spark-plug *D* after entering the combustion chamber, the flame cap will travel against the gas current until it reaches a point where the velocity is the same as the rate of flame propagation, in this case, 50 ft. a second.

Such a point must exist between the point where the velocity of the gases is zero and the point of entrance. Otherwise the flame will travel through the opening through which the gas is entering, and ignite the mixture approaching the burner. To confine the flame

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within the burner it is therefore necessary at all times to maintain at the point of entrance a velocity higher than that of the flame propagation. This will result in maintaining the flame within the burner and the products of combustion will issue from the outlet *E* of its combustion chamber.

In practical application of the burner already made it has been found that the burner can be operated over a wide range of pressure differences without adjustment and it has also been found possible to use the burner with the heaviest and cheapest grades of oil obtainable and still secure complete combustion, at least so far as the eye and nose can detect.

Use of Fuels of Low Volatility

In case liquid fuel of low volatility is used it is of course necessary to heat the air in which the fuel is mixed and see that the latter is finely divided. In practice this is done as follows:

The air, passing through the atomizing device *25* (Fig. 2), becomes impregnated with fuel mist and is immediately conducted through tubes where its temperature is raised by contact with the hot walls of the inner tubes *11*, carrying exhaust gases. This exhaust heating at constant pressure, not only effects the material increase in efficiency already noted, but serves to make a fixed gas of the mixture, which may thereafter be safely conducted to the point of combustion without fear of condensation. With gasoline this fixation is unnecessary. Hence a ready means for starting a cold engine is available.

In applying the burner to an engine it is necessary simply to see that the conditions outlined for properly mixing and volatilizing the fuel are met. If the temperature of the air passing the nozzle *10*, Fig. 2, is sufficiently high to cause immediate ignition of the fuel, two alternatives are open. The first is to maintain a velocity in the mixing chamber that is always greater than the rate of flame propagation in the mixture. The second is to make provision as by valves *23* and *24*, Fig. 2, whereby the mixture while on its way to the burner is too rich to ignite, that is, until sufficient air entering through auxiliary inlet *F*, Fig. 3, is added to this over-rich mixture in the space surrounding the combustion chamber so as to secure complete combustion.

The degree of rapidity at which the heat is liberated in the burner is indicated by the fact that it has been found possible in tests already made to melt a bar of steel inserted in a burner made of brass. The design of the burner is such that the gases entering insulate the walls so that the latter remain comparatively cool.

ADVANTAGES OF CYCLE AS APPLIED TO AUTOMOBILE ENGINES

The advantages resulting from the use in a motor vehicle of an engine operated on the proposed cycle include all of the inherent ad-

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vantages of the constant pressure cycle previously mentioned. While the field in which it can be applied is by no means limited to that of the motor vehicle this cycle is peculiarly adapted to motor-vehicle engines for the following reasons:

(1) It renders available for use fuels, such as kerosene and oil fuel, now produced in such large quantities (to supply the demand for gasoline) that they have become more or less of a drug on the market.

(2) It is thermally efficient at all loads because of the constant compression pressure, utilization of exhaust heat and other factors outlined under the heading "Advantages of the Constant Pressure Cycle."

(3) It produces a more uniform torque and is smoother in operation than any Otto cycle engine owing to the fact that fuel can be admitted through a considerable proportion of the stroke and that there is no sudden rise of pressure such as produces sudden and violent shocks in an engine operating on the Otto cycle. Under these conditions it is impossible to see any necessity for an engine of more than four cylinders, while it is probable one of fewer cylinders can readily be used.

(4) It has a considerable overload capacity. In other words the normal turning effort can be much increased either by increasing the pressure on the unloading valve and thus raising the compression pressure or by lengthening the admission period up to practically full stroke. Under these circumstances it is possible to dispense entirely with a change speed gear as is done in the case of steam-driven cars and locomotives.

(5) It can be operated on a two-stroke cycle and thus further decrease the variation in torque and the loss in power resulting from the two idle strokes in a four-stroke cycle engine. In short the engine will possess all the inherent advantages of the two-stroke as compared with the four-stroke cycle without any of the disadvantages of the former, which have operated against the success of a two-stroke Otto cycle engine.

(6) It can be built to produce higher mean effective pressures than do engines operating on the Otto cycle. For this reason and because of its adaptability to a two-stroke cycle the engine can be made much lighter for a given normal power than any Otto cycle engine practical for motor-vehicle use. The design can also be made as compact as that of the motor-vehicle engines now used.

(7) It will probably require no starting device. When the engine is stopped after running for even a short period of time, air under pressure will remain in the receiver. Under these conditions it will be necessary only to open the throttle, permit the air remaining under pressure to flow into the mixing chamber and thus through the burner into the cylinder. If the clutch be held out during this starting period the air pressure even though it be low will under normal con-

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ditions be sufficient to start the engine. Once started the engine will in a few turns fill the receiver with air at the normal compression pressure of the cycle. In case an engine stops on dead center and all air in the receiver leaks out, the engine can be cranked by hand or a small hand pump used to fill the receiver.

(8) It will in all probability be comparatively free from difficulties due to the accumulation of carbon, providing a reasonably good grade of lubricating oil is employed, because of the ideal combustion conditions previously mentioned.

(9) It will operate to decrease car weight on account of its own light weight and the fact that no gear change or starter will be required. Owing to the smooth torque characteristics the whole driving mechanism can probably be made of lighter weight than under present conditions. It is probable also that no muffler will be required on account of the expansion and cooling of the exhaust gases occurring in the piping where the transfer of heat to the incoming air takes place. It should be entirely free from difficulties corresponding to those now resulting from imperfections in carbureter design and operation.

(10) Its use will result in a much simpler control of the car owing to the absence of any gear-changing mechanism. There is no occasion for a spark advance mechanism since the mixture will always be ignited the moment it enters the burner; hence pre-ignition is impossible.

AUTHORS' CONCLUSION

In view of the numerous advantages of the constant pressure cycle and the peculiar advantages of the proposed theory of construction the authors anticipate a rapid change from the constant volume to the constant pressure type of engine in automobile construction.

BIBLIOGRAPHY

The authors have purposely avoided the use in this paper of formulas having to do with the thermodynamics of the constant pressure cycle, for the reason that this ground is covered thoroughly in most reference books on the gas engine. Those interested in a further study of this subject will find much valuable information in the list of books given below.

Particular attention is called to the paper by R. M. Neilson, quoted at length in Suplee's book on The Gas Turbine, in which the author states that with a compression pressure of only 30 lb. absolute (approximately 15 lb. gage) a constant pressure engine can have an *ideal* efficiency of 84 per cent when all the exhaust heat is returned to the charge. This is the highest ideal efficiency of any cycle mentioned, including the Otto cycle with an ideal efficiency of 40 to 45 per cent.

Another point worthy of special mention is that cited by Hiscox in his book entitled Compressed Air and Its Applications. In the chapter devoted to Compressed Air Reheating and Its Work it is

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made evident that one of the most efficient methods of converting heat energy into useful work is to heat air already compressed and expand it in an air motor. Hiscox states that 1.28 hp. can be developed by the combustion of 1 lb. of coal per hour, or where the internal combustion method of heating is used (products of combustion are added to the air) no less than 2.4 hp. per pound of coal per hour can be developed—a far better efficiency than can be realized in any known type of steam powerplant.

The constant pressure engine can be considered practically as being an air motor in which the air is heated after being compressed and then allowed to expand in the same cylinder in which compression takes place. In the cycle proposed in the paper heating is done in a most efficient manner with slight opportunity for loss of heat energy, since the latter is not liberated until the air enters the cylinder in which the work is done. Furthermore, a considerable proportion of the heat normally lost in the exhaust is returned to the next charge of air entering the cylinder.

REFERENCE BOOKS

The following books contain valuable information in regard to the constant pressure cycle:

Frederick Remsen Hutton, "The Gas Engine"; Carpenter and Diederichs, "Internal Combustion Engine"; Hugo Güldner, "Design and Construction of Internal Combustion Engines," translated by Herman Diederichs; Charles Edward Lucke, "Gas Engine Design"; Henry H. Suplee, "The Gas Turbine," and Gardner D. Hiscox, "Compressed Air and Its Applications."

DISCUSSION

H. P. MAXIM:—When the Otto-cycle engine is offered in units having no less than twelve separate cylinders, a better cycle becomes a matter of necessity. Engineers look at the question in two different lights. Those of us who must design and build something this month, which can be sold next month, regard anything other than the Otto cycle as partaking of the fanciful or hair-brained. On the other hand, those of us who are not actually dependent upon the immediate sale of the design, are keenly interested in anything that promises better thermal efficiency than the Otto cycle.

I feel that the Otto-cycle engine has succeeded in spite of and not on account of its principles. The tremendous losses at the exhaust, the losses due to the adulteration of the fresh charge by the large quantity of burned gas from the previous charge and those resulting from ignition at lowered compression, as when running throttled, cannot be continued long. The only way we can permit them is by having no other alternative. Messrs. Browne and Chase are unquestionably justified in returning to the Bray-

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ton idea. Since Brayton's day we have learned a vast amount. Everything from piston fit and crankpin-bearing design to flameless surface combustion has been tremendously improved since his day. The laws governing the combustion of gases are far better understood to-day than they were in Brayton's time.

I believe however that the authors go only part way when they pick up the old constant-pressure cycle and apply it to a multiple-cylinder reciprocating engine. I have been at work upon this cycle for some time, and the result of the study has been that a continuous rotary mechanical design seems to be possible. If a continuous rotary design running at extremely high speed is practicable, the possibilities become wonderful indeed. All of the saving of thermal energy claimed by Messrs. Browne and Chase will be made. A much lighter and more convenient piece of mechanism will be secured. As a result, whereas we now use 1 gal. of gasoline to accomplish a given effort, we can expect to use only a pint; while our mechanism now costs $100, it will cost only $50.

The automobile developed the light reciprocating gasoline engine. The successful development of this engine made the flying machine possible. Now the flying machine becomes a necessity of such immense importance that enough pressure is brought to bear to bring about the development of an entirely new gasoline engine. This will be accomplished within the next five years. Then will the aeroplane return the courtesy of the automobile by extending to the latter its new motive power.

POSSIBILITY OF GAS TURBINES

HERBERT CHASE:—The development of a turbine operating on the constant-pressure cycle has long been predicted. By utilizing the burner described in the paper such a turbine can doubtless be made to operate, providing the problems involved in cooling can be solved. The chief obstacle seems to be that of compressing the air necessary for combustion of the gas. We hear that some moderately efficient rotary air compressors have been developed in France, but have no thoroughly reliable data regarding them. If a reciprocating compressor must be employed, there seems to be no object in using a turbine in which to do the work of the cycle, since this can be done to better advantage in the compressor cylinder, as shown in the paper.

Granting however that an efficient turbine will be developed, it does not follow by any means that it will replace engines of the reciprocating type. In the steam-engine field, for example, the reciprocating type is not only more efficient in the smaller units but also more readily adaptable to most conditions of service. Unfortunately the turbine, whether steam or gasoline, is essentially a high-speed machine. On this account its application to motor-vehicle service, in which comparatively slow speeds and high torques are required, is not a simple problem. It is by no means certain that

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even a highly efficient gas turbine will, if developed, replace the reciprocating engine for this class of work.

A. L. RIKER:—I have had the constant-pressure-cycle engine in mind for many years, and have felt that it should receive some consideration. That this cycle can be applied to high-speed engines is doubtful; it is possible however that it can be used successfully with the low-speed type of engine for commercial vehicles.

HERBERT CHASE:—We realize that some difficulty may be experienced in producing an engine of the constant-pressure type that will run at the high speeds now popular in automobile construction. We feel however that the higher mean effective pressure and more regular torque characteristics of the constant-pressure type will make it unnecessary to use high speeds. It seems entirely possible to use lower speeds with little if any gear reduction, approximating closely the conditions that apply in steam-driven cars as now constructed.

JOHN WILKINSON:—I was familiar with the old Brayton engine used at Cornell University for many years. While I never remember seeing it running, it was a familiar sight. In 1900 I attempted to build an engine of this type. It was not any sort of a success, principally because we were not able to secure ignition satisfactorily at anything but low speeds. The Association of Licensed Automobile Manufacturers, in connection with the Selden patent, tried to build an engine of this type. Many thousands of dollars were spent.

MECHANICAL EFFICIENCY

I doubt the statement that the theoretical efficiency of the constant-pressure-cycle engine is as high as 84 per cent. In fact, as it would be used in an automobile, I doubt if the efficiency at full load would be as high as with our present engines, although perhaps it would be considerably higher at the light loads.

PROF. HERMAN DIEDERICHS:—Test results for the Brayton engine are quoted both by Clerk and G. Güldner. The main trouble appears to have been that the pump work was so large a proportion of the indicated horsepower that the mechanical efficiency was less than 50 per cent. That I suppose was partly inherent in the bad mechanical work of the period. I believe, however, that any type of small engine without outside compression in a separate pump will suffer under the same disadvantage to-day.

The Brayton engines did not compress in the power cylinder. That of course would not be permissible to-day as it makes for low thermal efficiency. We still have one of the Brayton engines in the Sibley Laboratory. It has been in operation during the last ten years. The engine described in the paper is, of course, operative, but I think that the efficiencies are problematical. One or two features need demonstration. This is particularly true of the oil-injector and the burner. The latter is, to my mind, the most important feature. An actual construction is the only thing that will prove this engine out completely.

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HERBERT CHASE:—It is exceedingly difficult to predict just what the thermal efficiency of the proposed engine will be. We fail to see however why the mechanical efficiency should be much, if any, lower than that of an Otto-cycle engine. We believe that the thermal efficiency should be much higher, partly because higher compression pressures and better conditions of combustion prevail, and partly because a good part of the heat that ordinarily would go out with the exhaust is utilized.

The burner described in the paper has, as stated, been found to operate successfully with widely varying pressure differences. No effort as yet has been made to apply it to an engine. The method of application and many other problems will have to be solved before an actual engine is produced. The oil-injector shown in the diagram was not intended to indicate actual construction; in fact the drawing of the engine is purely diagrammatic; just how the injector and other details will be worked out remains to be seen.

A. H. EHLE:—Considerable experimentation will be necessary prior to the evolution of an engine working as smoothly and certainly as the present highly developed automobile engines employing the Otto cycle. I feel that an engine of the kind proposed, with higher thermal efficiency and more flexible qualities, will almost surely be developed. Such questions as proportion, pressures and relation of parts seem to be the only matters requiring determination.

SCAVENGING ACTION

W. W. WELLS:—I am glad to note the prominence given in the paper to the subject of clearing the cylinder of burned gases. I do not see any reason however for the claim that the constant-pressure engine can be scavenged much more effectively than the Otto type.

I have often wondered why some automobile or engine builder does not develop a scavenging engine of either the two- or four-stroke type of constant-volume cycle. This seems to be the most promising field for engine development. The only experiments along this line that I have heard of were with a six-stroke cycle. The scavenging action of the two extra strokes showed some marked advantages, which were however not sufficient to outweigh the disadvantages of the six-stroke cycle.

In 1911 I secured a patent on a scavenging two-cycle engine in which a differential piston pumped air to scavenge the adjoining cylinder in the same manner as shown in Fig. 2. I also sketched some devices for utilizing a closed crankcase or a differential piston to pump air for scavenging a four-cycle engine and I still believe they are entirely practical ideas.

The scavenging action of the proposed engine is by no means complete, for even with a differential piston having a displacement greater than that of the working piston, the air would not expel all the burned gas from the corners of the cylinder and some would be pumped into the reservoir.

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A. P. BRUSH:—The Otto-cycle engine is a bridge that has carried us safely for a long time. We should not criticize it more severely than it deserves. Mr. Chase has referred to the sudden explosion of the Otto-cycle engine. In reality the sudden explosion is not a limitation because the maximum load on the parts of the high-speed type of engine does not come from the explosion pressure; the explosion pressure is much lower than the pressure set up by the inertia of non-uniform motion parts.

If I understand the cycle, zero clearance is proposed for the cylinder, or as nearly that as can be secured. As the air is compressed it will be forced into a receiver. From that receiver it will be drawn, mixed with fuel, heated to some extent by exhaust gases and to a further extent as it goes into the cylinder by the burning of the fuel and the oxygen of the air. The air must be compressed to a pressure higher than the pressure during combustion, to force the amount of air into the receiver; the amount that the compression pressure is above the maximum working pressure represents work done in forcing the air into the receiver, and of course reduces the area of the card. It seems to me that to reach the maximum theoretical efficiency of the proposed engine, a receiver pressure must be carried at least as high as we have found it possible to carry explosion pressures in the Otto-cycle engine; otherwise the peak of the card will be lost.

NITROGEN AN ASSET

I believe Mr. Chase said, "We cannot eliminate the nitrogen." We are apt to misunderstand the function of the nitrogen. It is perhaps proper to speak of the internal-combustion engine as a nitrogen engine rather than anything else. Nitrogen is really the medium through which we secure power, without having abnormal temperatures. What we do virtually in the Otto-cycle engine is to compress a charge of nitrogen mixed with oxygen, which is one-half of the combustible, and hydrocarbon, the other half; then we heat and really take the work from the nitrogen. We have some carbon dioxide and some water vapor, but relatively less power comes from the products of combustion than from the heating of the nitrogen in the charge. Pure oxygen and a gaseous hydrocarbon would cause explosions of enormous suddenness and high temperature; the nitrogen of course decreases the rate of flame propagation and holds down the temperature. In all internal-combustion engines we must consider the nitrogen an asset and not a liability.

HERBERT CHASE:—The Otto-cycle engine has been a most serviceable prime-mover for many years, and will, I have no doubt, be used extensively for many years. Nevertheless, it has many shortcomings. The greater of these we have endeavored to point out. Mr. Brush will, I am sure, agree that progress in the art can result only from recognizing the faults of what we have and striving to develop new engines that eliminate or have fewer faults.

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The bearing pressures due to explosion pressures are undoubtedly less than those due to inertia forces *in high-speed engines.* A lower maximum pressure on the piston-head will however permit the use of a lighter piston-head and decrease bearing wear by reducing the reciprocating mass and lowering the bearing pressure during part of the working stroke. Our remarks in this regard refer to slow-speed engines in which the inertia forces are unquestionably lower than those due to the explosion pressure.

Any mechanism subjected to hammer-blows is apt to deteriorate more rapidly than one in which the load is more gradually applied. The rapid rise of pressure that follows ignition in the Otto-cycle engine amounts practically to a hammer-blow and is far harder on the mechanism than the pressure of a constant-pressure cycle engine which does not rise suddenly at any time.

The card given in our paper, which is purely theoretical, indicates that the clearance is zero at the end of the stroke. This is practically true insofar as the clearance in the cylinder proper is concerned. In reality however the receiver forms the clearance space. The compressed air is delivered to the receiver and returned to the cylinder without decrease in pressure except that due to frictional resistance. Probably the actual card will resemble the theoretical card closely, except that expansion to atmosphere will occur only at light load.

When the engine operates however in such a way that compression starts at the point *A*, all of the air compressed in one stroke cannot be used in the working stroke without running a late cut-off. Of course it will not then be possible to expand to atmosphere before the end of the stroke, but the area of the card will be greater than that obtained with the Otto-cycle, provided the compression pressure is sufficiently high. In the case, however, of the constant-pressure cycle engine the compression pressure can be carried much higher than with the Otto-cycle on account of the fact that preignition cannot occur, since only air is compressed. Under these circumstances the horsepower of an engine of a given displacement can be made much greater than that of an engine operating on the Otto-cycle. With the higher compression pressure the efficiency of the engine will be greater even at late cut-off (that is, full load) than that of the Otto-cycle; whereas under part load it will be much higher. This is especially important in the case of automobile engines, which run during a large part of the time under less than half maximum load.

The reference made to nitrogen had to do only with consideration of the various classes of combustion, without specific reference to combustion conditions in an engine. The function of nitrogen is, as Mr. Brush indicates, a useful one, but it is none the less true that combustion proceeds more rapidly in an atmosphere free from inert gas such as nitrogen.

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COMPRESSED-AIR DEVICES

ARTHUR B. BROWNE:—The card of the constant-pressure cycle differs only in detail from the card of a steam engine. It is practically identical with the card of a compressed-air motor, the only exception being in the compression line.

In the application of the proposed cycle, the engine on its working strokes is simply a compressed-air motor. On the reverse stroke it is a simple air-compressor. If all the air compressed on one stroke were utilized on the return power-stroke, without addition of heat, the engine would be inoperative. The air is expanded by reheating, so that the volume available for the production of power is greater than that originally compressed. This increased volume is the measure of the useful work of the cycle.

A given power output requires a definite volume of air at a given pressure; hence, as the demand for power varies, as with a varying load, some means must be provided whereby a proportionately variable amount of air will be compressed, because the expansion ratio (by heat) remains practically constant at all loads. This is the purpose of the unloading device *15*, shown in Fig. 2 of the paper.

There is really little that is undemonstrated in the proposed application of the cycle. That the cycle itself is wholly practical has been proven by the hundreds of Brayton engines that were in actual use until they were replaced by the more compact design of the Otto cycle. The burner is an accomplished fact. It seems that the practical application of the proposed device is solely a matter of detailed mechanical design.

Compressed-air motors are in operation throughout the world, many of them using reheated air. The highly efficient use of fuel so applied is thoroughly proved by the records of motors used on the street railways of Paris. Air-compressors provided with automatic unloading devices are too well known to need further comment, while the efficiency of regenerative processes utilizing exhaust heat has been demonstrated repeatedly in European developments of the gas turbine.

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KEROSENE VERSUS GASOLINE IN AUTOMOBILE ENGINES

By Dr. Charles E. Lucke

(Member of the Society)

Abstract

The author outlines the factors leading to the present high cost of automobile fuel, states that the introduction of new distillation processes will not solve the problem, but that the development of kerosene-utilizing appliances will produce results satisfactory to everybody.

It is stated why kerosene cannot be used on the present gasoline cars. The adaptation of the gasoline automobile engine to the use of heavier fuels than will vaporize without the use of heat is entirely a problem of heating and heaters.

The author reviews at length the principles embodied in and the construction of the heated vaporizers or vaporizing heaters now used in stationary and traction kerosene engines and in alcohol engines, giving illustrations of a number of such devices.

After thus developing what in his opinion are desirable and good principles, the author describes a form of vaporizer embodying such principles, which he states has had successful trials (both block and road) in automobile service. A semi-automatic starting burner to accompany the vaporizer is also described, as regards both construction and operation. In conclusion the hope is expressed that the principles outlined will result in the production and use of kerosene automobiles on a scale sufficiently large to affect the price of fuel within the next year.

STAGES IN MACHINE DEVELOPMENT

Machinery of practically every class has developed in a series of stages, during any one of which there is a slow regular perfection and standardization of generally accepted and widely used types, each new stage being inaugurated by more or less radical changes in some part or in the whole. These latter constitute the milestones of progress, and are due to perfectly natural causes, such as changes in available materials, or in relative prices, scientific discoveries, inventions, new shop methods or in new economic conditions in general. In these days of large interests and big business, it is most important that this situation be recognized, because plans must be made more and more in advance, the bigger the project and the greater the interests involved. Especially important is it to be on the lookout for the coming of the periodic change, so that conditions of preparedness can be established that will prevent surprise and avoid the business chaos that would otherwise follow.

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The time to expect a change of this sort is the time when the old order seems to be most firmly established when viewed from the inside; in fact, so true is this that one might almost say that the older a thing in this field the most sure a change, and the expectation becomes a certainty when broad economic conditions surrounding its production and use have changed in the meantime. In the general power field the century-old reciprocating steam engine has felt the competition of the steam turbine and gas engine, and on land of the electrically transmitted water power. The production of oil and its refinings are responsible for the rise of the oil-fired boiler, the heavy oil engine and finally the gasoline engine, that mechanism of most surprising economic consequences. The automobile itself is a milestone product, and each element of it has likewise passed through the regular succession and alteration of radical changes and standardization.

While each big change is due to pressure of one or more of the kinds mentioned, it is itself the cause of another, and exerts a sort of reactionary influence on these conditions that requires constant study to even understand, much less control. Thus, the automobile, aided by the motor boat, the aeroplane and the farm engine, created by, and developed on the production of gasoline as a prime factor, is now responsible for such a strong reactionary influence on the oil refining industry as to warrant the feeling that a new milestone in its history has been reached, one requiring vigorous, concentrated and organized effort to meet properly.

CAUSE OF HIGH PRICE OF GASOLINE

Automobile designers, manufacturers and users have become accustomed to the idea that gasoline is the one proper and necessary fuel, and have consistently demanded it, rejecting everything else, except perhaps those few substitutes that have properties so similar as to require no change in the construction of the mechanism or in its use. This view, imposed on the oil refiner in the form of ever more rapidly increasing demands for gasoline production, has resulted in some reactionary consequences, natural enough in view of the market conditions that limit him in the purchase of crude, with the sale of all of its refinery products in addition to gasoline on the one hand, and the chemical limitations on the yield of these products on the other.

It is natural that the refiner, meeting yearly the demand for greatly increased quantities of gasoline, and finding himself saddled with a much larger corresponding yield of heavy distillates of the kerosene and gas-oil class without any similar demand, should be compelled to readjust his prices as a first step in saving himself from an unsalable congestion of heavy products, and from financial loss. It is also natural that the refiner should seek crudes that have high natural gasoline yields, but it necessarily follows that these

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crudes themselves should, as a consequence, rise in price, and so more or less neutralize the advantage of their use.

Thus the oil refiner, continuing to accept the demands of the automobile industry for more gasoline, and subjected to public criticism and condemnation of his gasoline-price increase, but also urged by a natural desire to put on the market just as much of this high-priced product as he can find or make, has finally been forced to undertake the development and introduction of new distillation processes of the cracking variety, and to rebuild refineries at large expense to work on the new systems.

The procedure of the refineries in developing new and higher gasoline-yielding methods is highly proper. The contributions of the oil chemists that have made it possible are most gratifying to all scientific men, but the results cannot be regarded as a satisfactory solution of the problem of cheap automobile fuel. In the first place, the action of the oil man is based on the assumption that gasoline is a necessity to the automobile; and that the demand for it is, and will continue to be, not only stable but more or less regularly increasing.

The oil man's interests are not served by the production of a cheap automobile fuel; he must have a satisfactory profit on a barrel of crude from the sale of all the products derived from it, and it matters little to him whether the average selling price is the result of a small yield of gasoline at high price with a large yield of heavier products at a lower price, as with the old standard cheap distillation processes; or of a more or less complete conversion of heavy products so that all salable material is gasoline at an intermediate price. As a matter of fact the best results all around are served not by additional refining costs such as the new cracking processes require, but by the use of the cheapest available refining methods, and the fullest possible use of all the products of that refining, the chief of which is kerosene. Development of satisfactory means of utilizing kerosene by automobile engineers will entirely change the situation from every standpoint, and produce results entirely satisfactory to everybody.

WHY CRACKING PROCESSES ARE UNSATISFACTORY

The new cracking processes do not constitute a satisfactory solution of the fuel problem for three principal reasons: (a) Kerosene cannot be removed from the market because it is an old commodity and the most perfectly distributed oil product in the world's markets; (b) the cost of installing new plants involves millions of dollars, which must be paid for by the new product, assuming the refiner can command the capital in the first place; (c) all of the new cracking processes are patented and patent rights certainly will not be made available to everybody, so that some re-

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fineries must continue as they are or be forced out of business by large interests commanding both capital and patent rights for the new processes and therefore in a position to keep up the prices of the products of these patent-protected processes.

Kerosene-utilizing appliances are more worthy of development than are the new oil cracking processes. Their perfection will constitute a truer and sounder solution of the present automobile-fuel price difficulty for the following leading reasons: (a) kerosene is now available in much larger quantities than gasoline, and no new refinery capital or patent rights are needed; (b) it is now sold at a much lower price, ranging from one-half to one quarter; (c) it is on sale in more places many times over, distributed throughout the whole world, including its wild places; (d) its use will, while increasing its price a little, also lower gasoline prices more, permitting the production of the latter in more refined grades for special purposes, resulting in a stabilizing of the prices in proportion as relative use corresponds to natural yield in cheaply operated standard stills.

That this situation is regarded of importance by the oil men, and that they are not only willing but anxious to cooperate in the development of the kerosene automobile is proved by the fact that the Independent Oil Men's Association appointed a committee, charged with this duty, only two months ago, as a result of the author's presenting before it a paper setting forth the possibility of success and the need for organization to perfect and bring into use such appliances. The kerosene automobile is not only a possibility but almost a reality, a reality now in the experimental sense, to become generally accepted and widely used just as soon as produced in sufficient quantities, mainly by new construction, but partly by old gasoline car conversion. The intermediate step is one of mechanical perfection of the kerosene-utilizing appliances with the least possible change in the rest of the engine and accessory parts, and it is this step that must be undertaken by the automobile engineers to warrant the support of the oil and the automobile manufacturing interests.

COOPERATION WILL DEVELOP KEROSENE CAR

The development of kerosene appliances adapted to the standard automobile engine with the least changes will be promoted by an acceptance of some guiding principles as standards of procedure, and it is hoped that the ideas laid down in this paper will serve this purpose. It must not be understood however that other methods or ideas are in any way belittled or condemned, or that some one may not be really better, but big results in an industry are not to be secured without cooperative effort of everybody concerned along lines generally acceptable as feasible and workable, even though they may not be best at first. If every automobile engineer approached this problem with the idea that he must invent something, and that his

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invention must be as different as possible from the other, the result would be chaos, not progress. Real progress, involving a satisfactory type of equipment in the shortest time with big scale adoption and use, big enough to affect prices of gasoline and kerosene simultaneously, will be secured only by cooperative development along similar, instead of dissimilar, lines based on sound principles. The final equipment must be capable of using not only kerosene but also gasoline, or, in general, any distillate however light or heavy, in which case neither prices nor products will matter. Then, and then only, will the automobile become independent of the oil market for refinery products, and the oil refining business can be conducted in the cheapest possible manner, producing the sound, economic condition of benefits to everybody concerned. The result must be reached quickly or the benefits to be derived will be so long deferred as to lose value thereby; quick results mean cooperative effort along similar lines, which unquestionably will produce satisfactory results.

The result of operating automobile engines on kerosene must, to be satisfactory, conform to certain requirements or specifications as to what such a result means. These are especially important in view of the fact that any engine when once heated up on gasoline can continue to run on kerosene after a fashion, a fact that has led many people to announce prematurely so-called solutions of the kerosene problem, and that is equally responsible for the often expressed opinion of users of these solutions, that "kerosene is no good," an unwarranted generalization. To be satisfactory, the results should be judged in terms of the trouble the user or operator may have as compared with gasoline troubles. In addition, the engine power or car speed, engine efficiency or car miles per gallon on kerosene, should be as near as possible to the corresponding figures for gasoline, but it is not necessary that these be equal if operating troubles have not been introduced. Reduction of engine power and car speed is easily compensated for by a slightly increased displacement obtained by adding a fraction of an inch to bore or stroke, without changing anything else; reduction of engine efficiency or car miles per gallons has its own compensation in the lower price of the kerosene.

TROUBLES WITH KEROSENE IN GASOLINE EQUIPMENT

Use of kerosene in standard gasoline equipment does not produce what can be regarded as satisfactory results from the operating standpoint because even in addition to the requirement of gasoline for starting purposes, the kerosene is so little vaporized as to involve troubles of the following well-known order: (a) Bad header distribution between carbureter and the several intakes, resulting in unequal charges to the several cylinders; (b) excessive washing down of lubricating oil from the cylinder walls, due to its solubility in kerosene—proved by the accumulation of kerosene in the crankcase oil; (c) smoke and smell in the exhaust, or internal

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carbon, due to decomposition of heavy unvaporized kerosene drops and wall films, or late vaporizing oil unmixed with air, by the explosion heat of the mixed part; (d) misfires and backfires, due to variations in the mixture as a result of varying degrees of vaporization of the oil that passes the carbureter as the engine temperature varies—especially noticeable with changes of throttle, engine speed or load.

The trouble encountered can be generalized as due to excessively wet mixtures, or, inversely stated, to incomplete vaporization and even to mixtures of a variable degree of vaporization. No particular knowledge of the properties of vapors or of vapor-air mixtures is required to realize that the sort of corrective needed is heat, but it makes all the difference in the world just how that heat is applied. The scientific knowledge of the properties of hydrocarbon liquids and vapors or their vapor-air mixtures, and of the laws of heat transmission through metal walls to get warm mixtures, is no more than sufficient to indicate just where, how and to what degree, the heat application is to be made.

The problem of adaptation of the gasoline automobile engine to the use of heavier fuels than will vaporize satisfactorily in air without the use of heat, is entirely a problem of heating, and heaters. Given suitable data, on the amount of heat required, on the temperature that should be maintained, on the design of suitable heaters in shape, arrangement and size, on the sources or available supplies of heat, on means of establishing and maintaining as long as necessary a suitable starting heat, on the connection between the heated mixture-making apparatus and the engine, and finally on the modifications required in the engine so that it will operate properly on the mixtures hot enough to burn properly without oil or residue, carbon deposits or smoke,—given such data, the kerosene automobile engine can be designed by any engineer. Of course, all this is not available at the present time, but enough is, to put quite satisfactory results within reach now, with the ordinary expectation of improvement, perfection and standardization of such equipment each year of successive use.

MIXTURE CONDITIONS CLASSIFIED

Mixture heaters or vaporizers have been much used in the related arts, with the stationary and traction engines using kerosene, and in the alcohol engine art, which was extended for a time in Europe to include the automobile field. Much information of value is derivable from these practices, but much must be done to secure equipment suitable for the light, flexible variable-speed throttle-controlled automobile engine confined to small space where stops are frequent and quick starts are necessary requirements.

These older practices are divisable into three groups, according to the condition of the mixture produced. A similar division of auto-

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mobile engine equipment is also possible with corresponding, but exaggerated, variations in operating conditions. These three mixture conditions are given the following names for the want of better ones: First, rain; second, fog; third, dry. The first of these is a wet rain-like mixture, hardly a mixture at all, in which much of the fuel is not only unvaporized but exists in the form of heavy raindrops, or stream-like films on the side walls of the passages. The second or fog condition is one in which much, if not most, of the oil is in the unvaporized state, but in drops so fine as to appear like tobacco smoke. These remain suspended in the air for a long time, as do the clouds in the atmosphere or moisture in steam but are equally subject to the collection and precipitation or separation as rain or wall films. The fog that has not precipitated as rain or wall film is reasonably well distributed through the air, making a true mixture as distinguished from the first or rain variety. The fuel is in the liquid rather than in the vapor state, so such mixtures can be quite cold as is also the case with the rainy mixture.

Dry mixtures, the third class, are perfectly clear and transparent, true invisible vapor and air being as thoroughly mixed, and as stable, as a gas and air mixture and as hot as the fuel properties require for the vapor state of so much fuel as is present with the air. Such mixtures are subject to no pipe or wall deposits, so long as the walls are warmer than the mixture, or the mixture is sufficiently superheated above its dew point, to use a useful term borrowed from the art of air conditioning with reference to water in dryers, evaporators and humidifiers.

ATMOSPHERE ILLUSTRATES MIXTURE CONDITIONS

The atmosphere with its water vapor constantly illustrates all three conditions, the thermodynamic laws of air conditioning, now fairly well developed, being precisely the ones required for controlling the mixture condition of air with kerosene vapor. The dry mixture of less than 100 per cent humidity is the ordinary invisible condition of the atmosphere. The 100 per cent humidity or saturation condition is a division zone between the dry and the fog mixture and is defined by a temperature and an air-to-water weight relation. The passage from one to the other is partly one of temperature and partly one of physical contact or intimacy. Passage from fog or dew to rain condition is promoted by time, by contact with solid surfaces and by sudden changes of direction of flow.

A dry mixture must be above a certain temperature called according to one point of view the drying temperature, and to another the dew point. This temperature is fixed by (a) the vapor pressure of the liquid as related to its temperature; (b) the relative weights of air and vapor in the mixture; and (c) the intimacy, uniformity or homogeneity of the mixture. In the case of the engine the weight relations or proportions of mixture constituents, air to vapor, are fixed

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by the carbureter, so that the remaining variables are the dryness, the temperature and the mixture homogeneity, or the temperature alone for a given design of mixture chamber. Above this temperature mixtures remain dry, and can be said to be superheated or to have a given humidity; below this temperature mixtures must be wet, but so much as is not liquid constitutes a saturated mixture, the free liquid being determined by the difference between the vapor-carrying capacity of air at the dryness temperature, and at the actual temperature, if these factors are known, as is the case for water vapor and air, though not yet for complex things like gasoline and kerosene.

FLOW OF WET MIXTURES

Wet mixtures, whether in the rainy or the fog condition, will, as they pass through the intake or manifold, produce a wall film of liquid, which flows in a way different from the air, the vapor or the fine fog, the difference being responsible for bad distribution to the branches. Such a wall film always moves more slowly than the gaseous stream and only moves at all because of friction between the liquid film and the true mixture; it distributes circumferentially around a passage in an unequal way whenever, as is always the case, the center stream itself is moving irregularly. For example, at a damper or butterfly throttle-valve the mixture is divided into two high-velocity streams, moving most rapidly along two lines at opposite ends of a pipe diameter at right angles to the throttle spindle. The liquid is thus forced around the circumference to the points where the vapor velocity is least, that is, in line with the spindle. Sighting through a glass tube the two streams of liquid would be visible, separated by two dry streaks. At an elbow the liquid can be seen in a glass tube to collect on the *inside* of the bend where the mixture velocity is least, and at a branch the liquid will pass from side to side of the feeder and branches as mixture velocity or flow direction changes, never dividing equally because of the time lag in the changes.

A decrease of mixture flow velocity causes greater accumulation of liquid, but a sudden increase tends to lift off the accumulation from the walls, drawing it through the engine with the effect of momentarily great excess of fuel, which may cause misfire and the action known as "choking." Again, such a film tends to produce a similar enriching effect whenever the throttle is closed suddenly, especially if it is somewhat warm, because of the sudden and great reduction of pressure, which can exceed 25 in. of mercury vacuum, and produces a sudden vaporization of the liquid film. Such conditions constitute the first objection to wet mixtures, especially to the rainy form, and to some, although to much less, extent to the fog form.

Single-cylinder engines suffer but little from these rainy mixtures because the whole product passes through one valve, and only the

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throttle movement producing film vaporization on closure, and liquid lifting on opening, remains. The mingled streams of mixture and liquid will partly vaporize and partly be sprayed on passing through the inlet valve. If the interior walls are hot enough, the liquid that strikes will be largely evaporated, while that remaining suspended from the inlet valve spraying will vaporize on compression. Cold, wet, rainy mixtures can therefore be used in such single-cylinder engines with fair success, or in multi-cylinder engines fitted with a separate carbureter for each cylinder, if the combustion chamber walls are kept hot enough; this is the method in use in standard farm tractor engines such as the Hart-Parr, International Harvester and Rumely. Of course, the mixture is not quite homogeneous, nor is it feasible to keep the walls hot enough, so some smoke may be expected at times and some lubricating oil solution or weeping, both being reduced as the walls become hot.

EFFECT OF WATER INJECTION

In order to carry sufficient compression to secure high mean effective pressures and thermal efficiency, or avoid the reduction of those accepted as standard for gasoline, water injection is resorted to and is successful. Its effect is threefold; first, to prevent the mixture in the cylinder reaching its ignition temperature too soon, by reason of the mixture cooling incidental to water evaporation during compression; second, to slightly raise the ignition temperature by the neutral steam dilution, also by such dilution to reduce the rate of propagation and produce a slow combustion instead of a detonating shock; and third, by water gas reaction of steam with nascent glowing free carbon in process of combustion in regions where the air is insufficient, to keep the interior clean.

Multicylinder automobile engines cannot use water injection, nor is it feasible to equip each cylinder with its own carbureter, though both of these things might be done if there were no other way; so it can be said that not only is the rainy, wet mixture unsuited to the automobile engine, but so also is the standard tractor engine modification of hot breech-end, with water injection and individual fuel supplies.

PRODUCTION OF FOG MIXTURES

Fog mixtures are less difficult to handle than the rainy variety, but require some special means for their production. Such mixtures exist at temperatures even lower than those of the present normal gasoline mixtures and can therefore be compressed as high or higher than gasoline as now used, with a correspondingly high mean effective pressure and large power output, and without water injection. It must be noted, however, that the efficiency is not correspondingly high, proving the high mean effective pressure to be the result of a cool dense charge rather than of efficient combustion, a fact borne

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out by the tendency to "weep," measured by kerosene accumulation in crankcase oil. Such mixtures give some trouble in distribution, due to a tendency to precipitate and form wall films, with the evil effects of such films already noted. So far but little progress has been made toward the development and use of uniform fog mixtures of this sort; so while this can be regarded as a possible line of de-

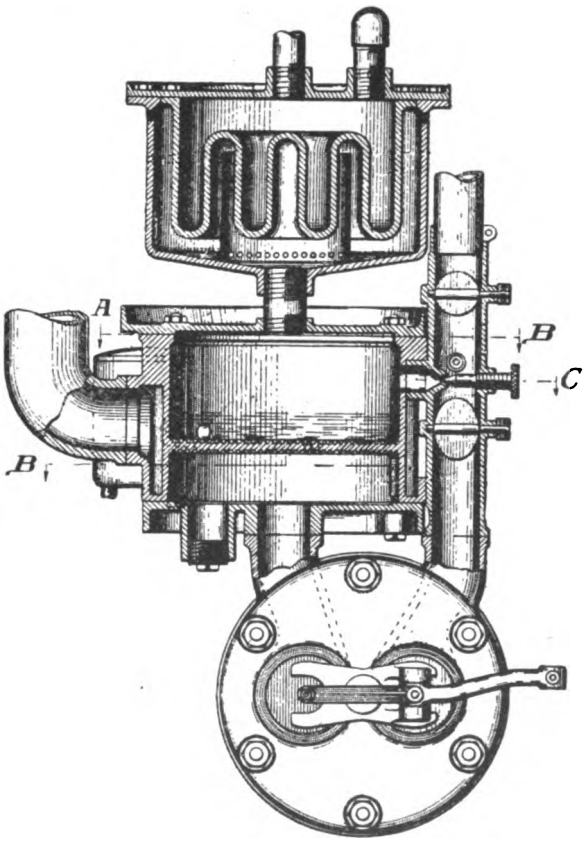

FIG. 1—LUCKE-VERPLANCK FOG-MIXTURE APPARATUS

velopment for the future, it must be put in second place at the present time. The dry mixture has so many advantages of its own and the means for its production are so much more fully developed, that it must be given the preference.

Before leaving the subject of the fog mixture, it is worth while to note two different ways of producing it, involving a common, final

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principle. This principle is, that a fine, persisting fog of oil in air, fine enough to float suspended for a considerable time and likewise to burn as well as a true vapor-air mixture, can be produced only by condensation of a hot dry vapor in its air, the condensation being effected best by bringing the hot vapor into contact with cold air, under conditions that produce a mixture. One way of doing this, that of Lucke and Ver Planck, developed some years ago and used on some stationary and tractor engines, is illustrated in Fig. 1. Here the exhaust is used to boil kerosene kept at constant level in a boiler-vaporizer, over which is fitted a water condenser constantly open to the atmosphere. Any excess heat over that required for the vapor used results in the production of excess vapor without rise of temperature, the excess vapor rising to the condenser, there giving up its heat to the water, the condensate dropping back to the boiler. In spite of any variations of heat supply, this apparatus provides a

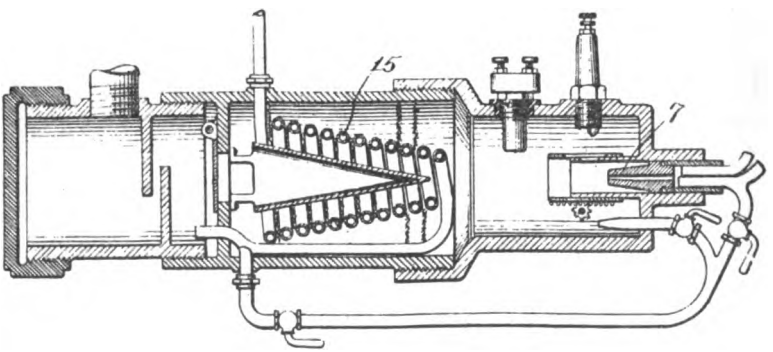

FIG. 2—SECTIONAL VIEW OF PORTER-RIDER FOG-MIXTURE APPARATUS

constant supply of vapor. The vaporization conditions are such as to involve absolutely no deposits of carbon or tar. The vapor is drawn from the boiler into the cold air stream entering the mixture pipe at the needle-valve *C* and beyond this point a reasonably cool air and fog mixture is produced in constant proportions, regardless of total quantity. Such an apparatus as this, while working well, seems to be unsuited for automobile use because of its bulk and the considerable time required for starting.

Another form of fog maker, that of Porter and Rider, Fig. 2, is of less bulk, but is lacking in means for keeping the air and fuel in constant proportions, while varying the total quantity. Oil is sprayed by compressed air from the nozzle *7* and an electric spark ignites the spray, causing some of the oil to burn in the limited amount of air present and vaporizing the rest. The flame is extinguished by passage between the coils of the cooling coil *15*, the

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hot vapor with the small amount of products of combustion mixing later with a main stream of cold air, where the cool fog is formed by condensation as in the previous case. There are other ways of forming these persisting fog mixtures, often called erroneously vapors, but for reasons noted they will not be mentioned, the two shown illustrating the idea that must be put aside at the present for the more favorable case of the dry mixture.

ADVANTAGES OF DRY MIXTURES

Dry mixtures have so many advantages and so few disadvantages as to warrant the belief that it is along these lines that effort can be concentrated with prospects not only for immediate success, but for success of a kind that will be followed by widespread general adoption of dry mixture apparatus. Such mixtures can be supplied to any number of cylinders through a straight header pipe with right-angled branches, and each cylinder will receive exactly the same charge as every other, regardless of load, speed or throttle, and without any special bulky manifold construction. In burning, the combustion is complete and perfect, assuming, of course, that a good carbureter is used to control the proportions of air to oil and keep the ratio constant. Smoke and carbon deposits are eliminated and so also is cylinder weeping. Such mixtures are hot, necessarily as hot as the fuel vapor pressure requires, and therefore the compression must be reduced, reducing horsepower and efficiency correspondingly but not to a serious degree. As everything depends on the temperature required, it is necessary to investigate this by considering the properties of the fuel and those of the mixture of vapor and fuel in combining proportions.

Fuel properties and mixture properties that are of importance to this question are (a) the composition of the fuel as indicated by the fractional distillation and ultimate analysis; (b) the vapor pressure-temperature curve of the fuel and of its several fractions and constituents; (c) the density of the constituent vapors and the mean density. Besides these properties a relation is required between the partial and total weights of constituents of the mixture and their corresponding pressures, as a tie or bond between proportions and dryness temperature for any one fuel.

It is well known for thermodynamic reasons that the relation between partial pressures of the air P_a and of a vapor P_v in a mixture; the ratio of air weight w_a to vapor weight w_v; and the density of air d_a to that of the vapor d_v in pounds per cubic foot, or the respective molecular weights m_a and m_v, is given by

$$\frac{P_v}{P_a}=\left(\frac{w_v}{w_a}\right)\times\left(\frac{d_a}{d_v}\right)=\left(\frac{w_v}{w_a}\right)\times\left(\frac{m_a}{m_v}\right)$$

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If such a mixture exists at a total mixture pressure of one atmosphere, the standard value of which is 760 mm. Hg., then

$$P_a = 760 - P_v$$

$$P_v = (760 - P_v)\left(\frac{w_v}{w_a}\right)\times\left(\frac{d_a}{d_v}\right)$$

$$P_v\left(1 + \frac{w_v}{w_a}\frac{d_a}{d_v}\right) = 760\frac{w_v}{w_a}\frac{d_a}{d_v}$$

whence, putting v_a and v_v as the volumes of air and vapor, or v_a/v_v as the cubic feet of air per cubic foot of vapor,

$$P_v = \frac{760}{1 + \left(\frac{w_a}{w_v}\frac{d_a}{d_v}\right)} = \frac{760}{1 + \frac{v_a}{v_v}} \qquad (2)$$

The result in (2) gives the pressure the vapor must exert alone in a mixture existing at 760 mm. Hg., in the weight proportions of w_a/w_v lb. of air per pound of fuel, when the vapor density with reference

TABLE I—DRYNESS VAPOR PRESSURE (MM. HG.) OF FUEL CONSTITUENTS

Substance	Temp in Deg. F. for 760 Mm. Hg. Vapor Pressure	Vapor Pressure Mm. Hg.
Pentane	95	19.2
Hexane	155	16.2
Heptane	208	14.0
Octane	257	12.5
Nonane	266	11.1
Decane	322	10.0
Undecane	382	9.2
Dodecane	418	8.4
Tredecane	453	7.8
Tetradecane	486	7.2
Pentadecane	518	6.7
Hexadecane	549	6.4
Heptadecane	577	6.0
Octadecane	603	5.6
Xenzine	177	20.4
Bolulene	232	17.1
Tylene	279	14.7
Methyl alcohol	148	92.3
Ethyl alcohol	173	49.5

to air is d_a/d_v, or when there is v_a/v_v cu ft. of air per cubic foot of vapor. From a vapor pressure-temperature table or curve for the substance, the temperature required to produce this vapor pressure and for such a mixture to be just dry can be obtained.

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It is evident from the form of (2) that this precise relation holds only for definite chemical compounds, because only such have definite vapor pressures and vapor densities. All other sub-

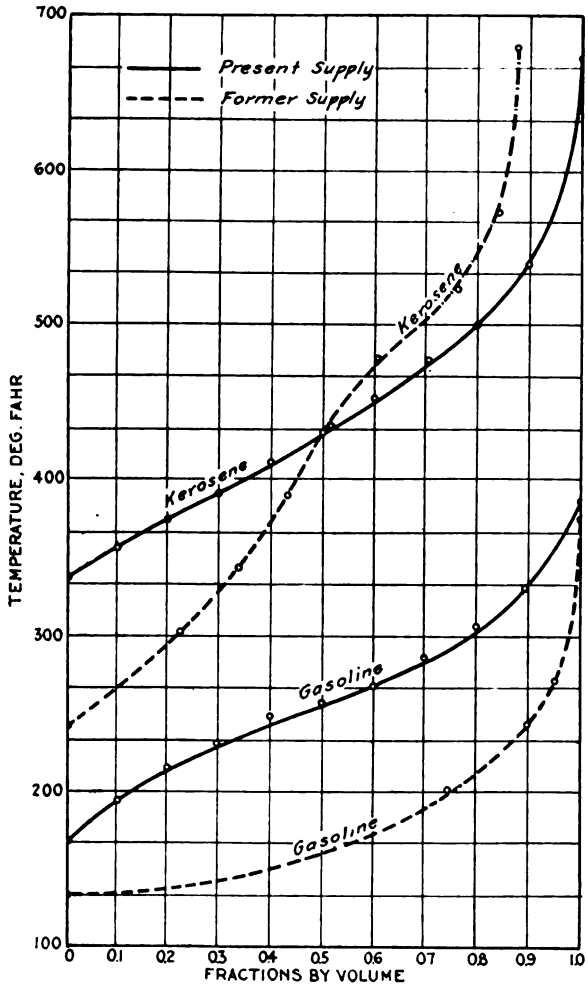

FIG. 3—TEMPERATURES OF GASOLINE AND KEROSENE DURING DISTILLATION

stances, such as the solutions of hydrocarbons in each other, as make up gasolines and kerosenes, must be treated in a roundabout fashion until new data become available. In Table I values are given for some paraffin hydrocarbons that are included in these fuels. Values for alcohols are added for the purpose of comparison.

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RELATION OF VAPOR PRESSURES AND OF TEMPERATURES

Reference to a vapor pressure curve, that of ethyl alcohol, for example, shows that a temperature of 72 deg. F. is necessary to develop the vapor pressure of 49 mm. Hg. required for dryness in the

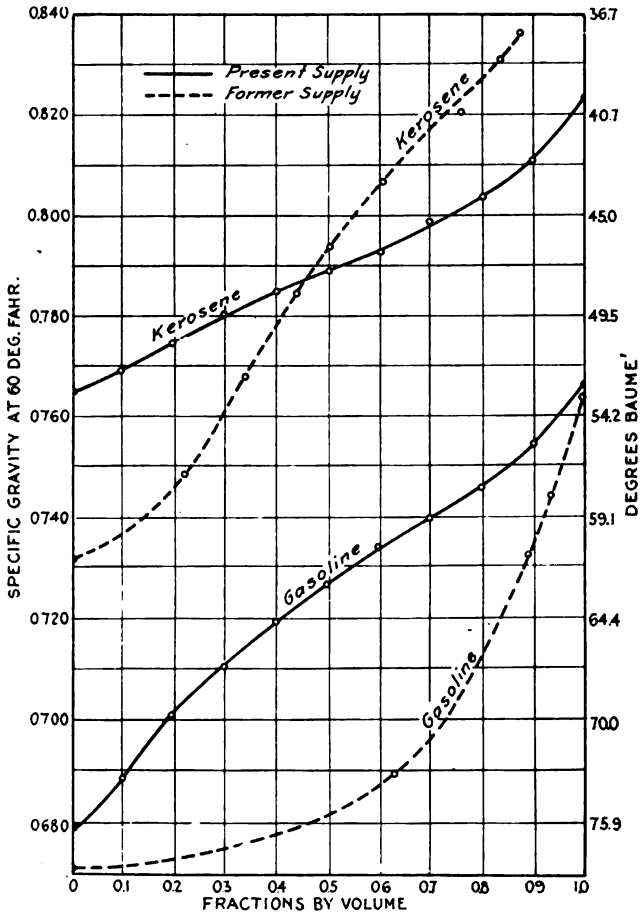

FIG. 4—DENSITIES OF GASOLINE AND KEROSENE DURING DISTILLATION

air mixture of combining proportions, and similar reference to the curve for hexane, which requires 16.2 mm. Hg. vapor pressure, shows that the temperature required is zero deg. F. approximately. At any higher temperature than these the mixtures would be superheated or the humidity less than 100 per cent, and condensation

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would be more remote. As the saturated condition is approached, the evaporation becomes more and more slow; so, as only a limited time is available in engine apparatus, an excess of temperature is desirable even temporarily. Also, a local depression of mixture pressure is desirable, because, according to the equation, if even for a moment this be half the standard 760 mm., then only half the vapor pressure need be developed, and if the mixture is just about hot

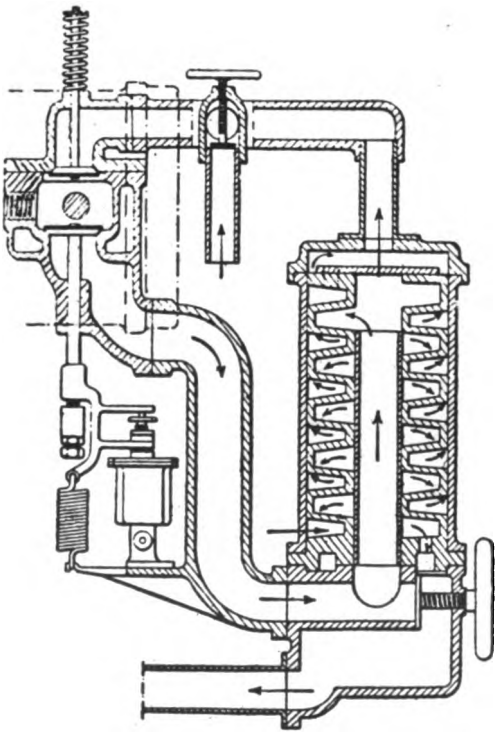

FIG. 5—BROUHOT (ALCOHOL) MIXTURE VAPORIZER

enough for full pressure, such a pressure depression will promote vaporization and leave the mixture saturated on recovery of the full pressure in shorter time, than without such a pressure depression or temperature excess.

Real fuels are not such simple compounds as these, but are solutions of one group of compounds in another, which can be partly though never wholly separated by fractional distillation, each fraction generally being one-tenth by volume of the original product. In Fig. 3 a present-day gasoline and a kerosene are given as fractionally dis-

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tilled. An older kerosene and a gasoline are shown dotted so that the changes in these classes of fuel that shortage of supply has forced, are made clear. In spite of higher prices, the present gasolines are

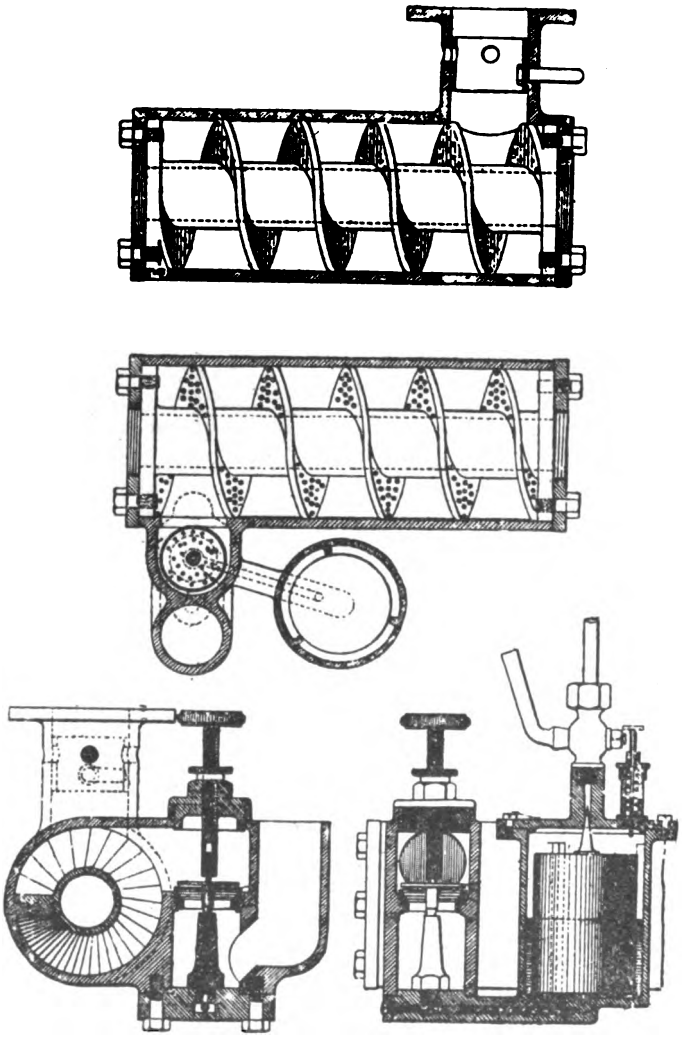

FIG. 6—MARTHA MIXTURE VAPORIZER OF HELICAL-COIL CLASS

heavier and much less volatile, the final temperatures of their several fractions rising more and more toward the values more characteristic of old kerosenes. The corresponding densities of the fractions are given in Fig. 4.

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The effect of these complex mixtures on the vapor pressure condition cannot be shown by a vapor-pressure curve for the fuel itself, and becomes clear only from a group of vapor pressure curves, one for each typical fraction. The most volatile fraction will be expressed by a curve of highest vapor pressures for the successive temperatures, while the least volatile will show small vapor pressures even at high temperatures. If one constituent did not interfere with the vaporization of another, as it really does, the relative vapor pressures at any temperature would be a measure of the relative volumes of the several vapors, and for complete vaporization

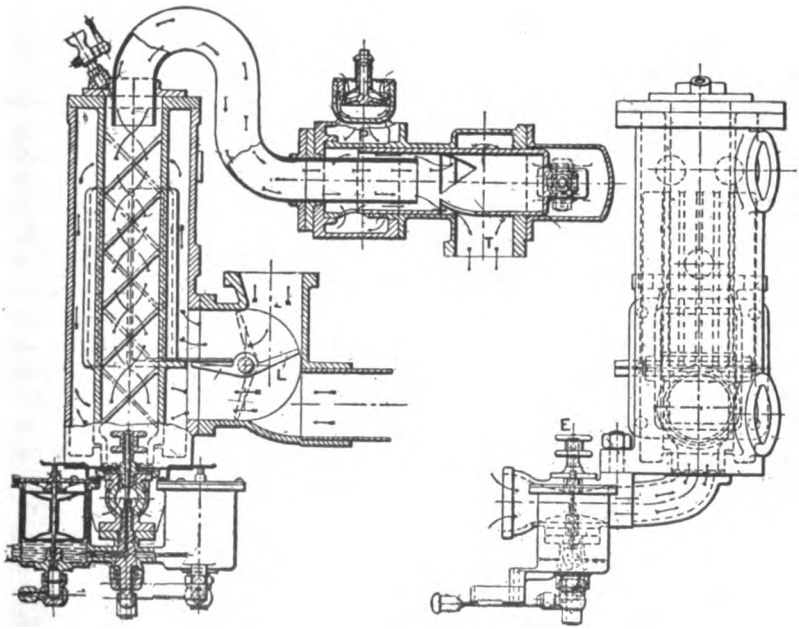

FIG. 7—THORNYCROFT MIXTURE VAPORIZER FOR KEROSENE EQUIPMENT

these should agree in weight equivalents with the proportions of constituents present, shown by the fraction curves.

At the present state of knowledge on this subject it is not possible to make any predictions of the sort desired, for dryness conditions similar to what are possible for simple compounds, but recent investigations indicate that there is a good chance that factors can be derived that will permit the same method to be used, starting with the fraction curves and the vapor pressure curves of the fractions separately. For such work the fractions should not be equal but divided according to uniformity of temperature range for any one.

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The average engineer would prefer to determine experimentally the dryness and dryness temperature by sight observation of a glass tube fitted with a thermometer, through which the mixture passes. Such sight tubes must be guarded with wire screens to avoid danger from accidental backfires.

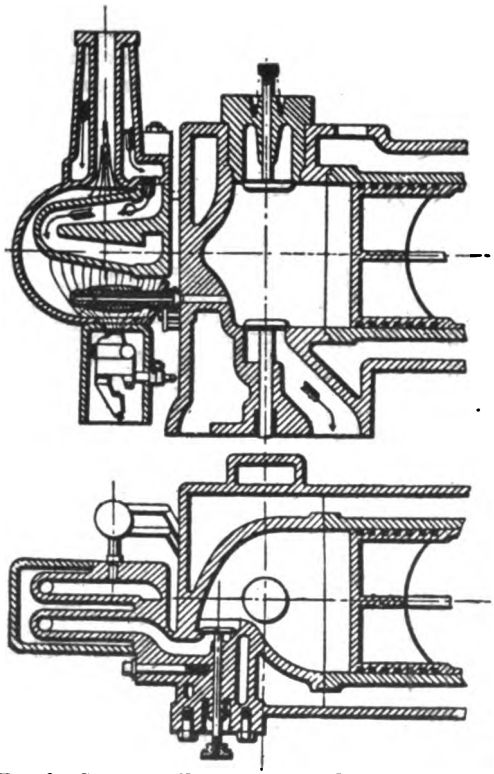

FIG. 8—CROSSLEY VAPORIZER FOR KEROSENE ENGINES

METHOD OF DETERMINING DRYING TEMPERATURES

The calculations in (1) and (2) have their special value in showing from the fuel properties alone whether one sample requires higher or lower temperatures to dry it than another. The vapor pressures of the fractions at a given temperature give an idea of the relative content of the vapors of each fraction that would come off at that temperature and are a good indication of the tendency of the heavier parts to come off last. This is most important because it is the basis of the deposits that are common to many

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vaporizers, and in it is to be found the reason for the vast difference between the actual temperatures required to dry the same mixture in one vaporizer as compared with another. If in one vaporizer or mixture heater there were the same degree of intimacy between the air and the vapor as in another, and if this were complete, the temperatures computed from the vapor pressure relations would be the ones sought and the same for all. Any variation in the degree of homogeneity of the mixture or any tendency for vapor to be in excess in one part over another, owing to vaporizer construction and flow conditions through it, will at once require a temperature in excess of the true mixture drying temperature, because when the vapor is in excess the weight relations are disturbed, and high vapor pressures must exist with corresponding higher temperatures.

A glance at (2) shows that as the ratio of vapor weight to air weight at any given part of the mixture rises, so must the vapor pressure rise. At the limit, for example, it might be imagined that there was no air at all over the surface of some liquid lying in a detached pocket. In this case the term $(w_a \div w_v) = 0$, and it follows that the vapor must reach a pressure not of a few millimeters of mercury, but the full atmospheric pressure of 760 mm. Hg. Now, this is most serious, and destructively harmful, because the heaviest constituents of the fuel cannot develop a full atmosphere of vapor pressure at any temperature whatever; before they even approach the necessary temperature, they are destroyed by the heat, breaking down first to tars and then to carbon coke. The same reasoning leads to the conclusion that none of the air should be by-passed around the vaporizer if the vaporization is to be most complete at the lowest temperature and in the shortest time, and, of course, vaporization must be accompanied by simultaneous air scrubbing and mixing.

One of the most important lessons to be learned from the analysis, therefore, is the basic and fundamental importance of most thorough and vigorous mixing of all the air with all the oil at the time of and during vaporization in order to secure the dryness condition at the lowest possible temperature. In this way, not only will the dangers of tar and carbon deposits be removed even with heavier distillates than kerosene, but also the dry mixtures will be the coolest, and the dryness temperature excess be the least, and the latter possibly zero. Cool mixtures are most desirable, but only when warm enough to be dry or substantially so, because they permit of maximum compression consistent with properly clean combustion. Their production by properly designed heated vaporizers requires the minimum of heat the lower the temperature.

Cases have been observed in which, owing to bad design of such vaporizers, mixtures above 500 deg. F. have not been dried, while with good design and the same kerosene fuel, perfect dryness was secured at 250 deg. F., half as much, and with a vast difference from the operating standpoint as well as from that of design. Not

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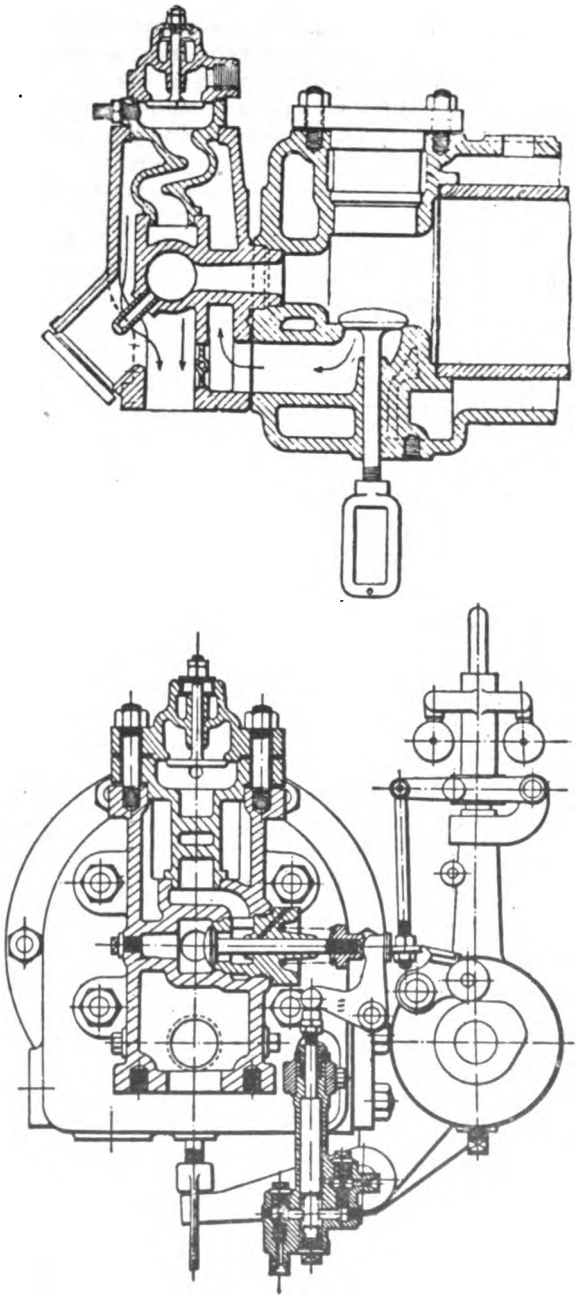

FIG. 9—FIELDING AND PLATT VAPORIZER WITH CROOKED, FREE PASSAGES

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only is the latter entirely feasible without pre-ignitions with reasonable compressions, but it is easily obtainable from the exhaust heat even when idling, whereas the former is not. The size of the vaporizer in terms of the surface required to transmit the quantity of heat needed with the mean temperature difference available, is much

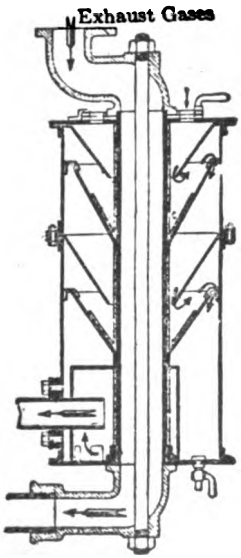

FIG. 10—PETREANO SEPARATING MIXTURE VAPORIZER

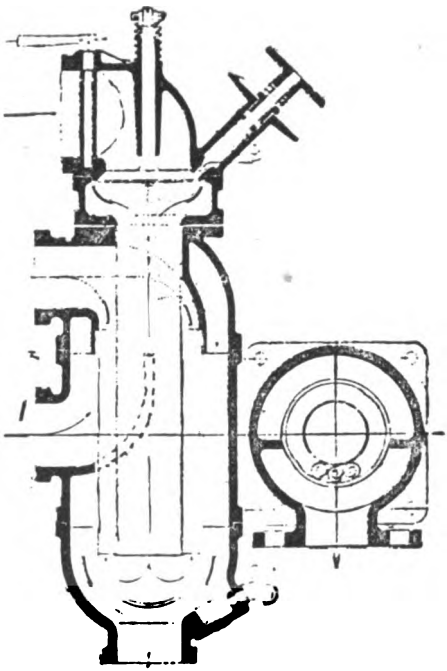

FIG. 11—DÜRR SEPARATING MIXTURE VAPORIZER

less than half when dryness is secured at 250 deg. F., because not only is the heat required about half, but the mean temperature difference is more than twice the former value.

CLASSIFICATION OF HEATED VAPORIZERS

In form and arrangement heated vaporizers are divisible into classes based on the sort of passage provided for the mixture during its heating and vaporizing period. Into the first class fall all those passages that are more or less equivalent to a helical coil. Among these are the Brouhot, Fig. 5 (Sorel), and Martha, Fig. 6 (Sidersky), old French alcohol arrangements, and the Thornycroft, Fig. 7 (Clerk and Burls), a more recent English kerosene equipment. The second

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class is characterized by crooked, free passages, somewhat equivalent to coils but of irregular rather than regular form, illustrated by Crossley, Fig. 8, and Fielding and Platt, Fig. 9 (Clerk and Burls), English kerosene oil engine vaporizers intended for stationary use and with hit-and-miss regulation.

Passages arranged to deflect the stream so as to separate out heavy liquid drops, and deposit the separated liquid into more or less isolated pockets in which vaporization takes place while the main stream passes by, are typical of the third class. The class is illustrated by the Petreano, Fig. 10 (Güldner), an old Italian form of multiple pockets, the Dürr, Fig. 11 (Sidersky), and Koerting, Fig. 12 (Güldner), both

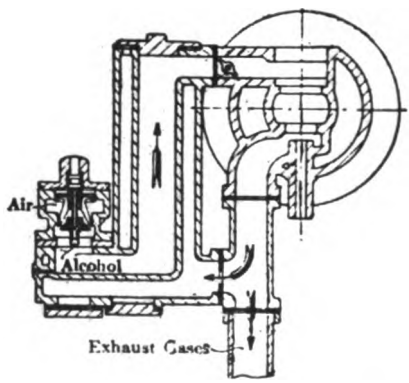

FIG. 12—KOERTING ALCOHOL MIXTURE VAPORIZER

German alcohol forms, the latter being of single liquid pocket type.

The fourth class includes all those forms of simple free passages that tend to force the wall film formed by the heavy liquid drops along the heated walls without depositing in pockets. This class is illustrated by the Campbell, Fig. 13 (Clerk and Burls), used with an English, and by the Capitaine, Fig. 14 (Clerk and Burls), with a Belgian oil engine. In the former an elbow, and in the latter the contracting walls of a straight conical passage serve to promote the formation of the wall film, and flow velocity tends to drive it along the hot surface where it is vaporized.

DIFFERENCES IN DETAILS

The several examples of each class differ somewhat in details one from another. In the first class, for example, the exhaust passes through a central pipe in the Martha, but in the Thornycroft it is outside, while in the Brouhot it first passes through the center and then returns by way of a counter helix. The last two stand vertically and

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the mixture flows upward as it vaporizes by its passage through the helix, while the former lies horizontally and its helix is perforated so that part of the mixture can pass directly through. In all three the primary object of the helix is to secure large heating surface for the mixture in a small space. The Thornycroft supplements this by longitudinally radial ribs extending into the exhaust passages, as exhaust heat is relied upon for operation. Both examples of the second class form part of the engine head. Exhaust heat is used in the Fielding and Platt but is supplemented by internal heat of com-

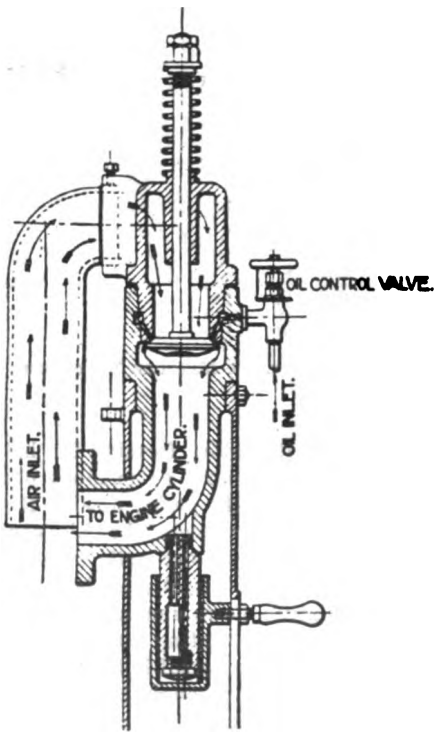

FIG. 13—CAMPBELL VAPORIZER FOR OIL ENGINE

bustion; part of the combustion chamber wall is also a part of the vaporizer wall. The Crossley relies directly on the heat of a blow torch. In both cases the passages are tortuous to fit available space and to give extended heating surface in a compact casting.

The separator-pocket (third class) forms are heated by a simple external exhaust jacket in the Koerting and Dürr, and by a central

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tube in the Petreano, the two pocket walls constituting inward extensions of the heating surface. Mixture flow in the last is alternately down toward the center and up toward the outside; the sharp angle pockets are next to the central exhaust wall where the liquid can lie and distil, practically excluded from the air stream, which takes the shortest path across the top, as indicated by the arrow. That flooding is expected is indicated by a bottom chamber below the outlet and by a drain-cock provided. Mixture descends in the Dürr through a large, long and straight central pipe and heavy liquid is deposited in a

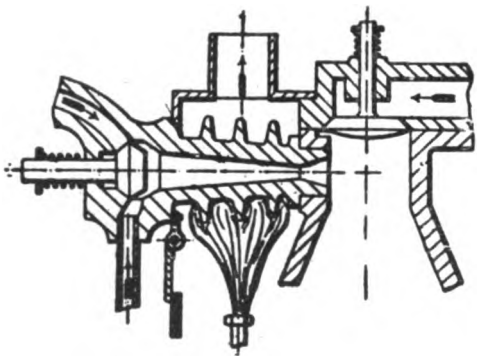

FIG. 14—CAPITAINE MIXTURE VAPORIZER

hemispherical pocket below its outlet, forming the bottom wall of the vaporizer and its least heated part, as exhaust passing down and out will not sweep the bottom of the pocket at all. It is on its upward return passage that heating is expected in this vaporizer, for here the mixture directly sweeps the exhaust walls with inward extensions, and it is this upper part that is directly heated by the exhaust. In general direction, the mixture flows similarly in the Koerting, first downward with heavy drops separating out, then slowly across and upward at low velocity, but the reversal is less abrupt and the bottom the most heated part, though not strongly heated because it is located in a sort of exhaust dead-end, the degree of heating being controlled by changing flanges and diaphragms.

In every device except the Petreano, and in part the Dürr, these heated surfaces are made of cast iron and all are extremely heavy, indicating that no importance is placed on quick starting, because initial heating time is, other things being equal, directly proportional to the weight of metal to be heated. Cast iron is, moreover, a necessary material for complicated passages, which must be formed by casting, and it is a natural selection for parts intended to be operated normally at a red heat, as is the case in nearly all of these. Red heats

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are necessary when kerosene is to be vaporized, with only a small part of the full amount of air, though not at all required when all the air passes through the vaporizer in contact with the oil, or when alcohol is to be vaporized. The importance of this air factor in required temperature is not recognized in some of these constructions, as is indicated by the passage of only a part of the air through the Brouhot, Thornycroft, Crossley, Fielding and Platt and Capitaine, while a full air passage is provided in the Martha, Petreano, Dürr, Koerting and Campbell vaporizers.

CONTROL OF AIR-FUEL RATIO IN VAPORIZERS

The proportionality between air and fuel at the constant value required for combustion that is necessary not only for best vaporization conditions, but even more for the proper combustion of these heavy vapors, is controlled in still more various ways. The one best arrangement, that of the ordinary gasoline carbureter, of particular value when all the air is in contact with the vaporizing liquid, is used in the Martha, Brouhot and Thornycroft; mechanical means of doubtful value for throttle-governed engines because of variations in feed are used in the Dürr, Koerting and Campbell, but in the hit-and-miss governed engines where there is no graduation of feed, it is perfectly well suited, because here a constant feed per stroke can easily be fixed mechanically with precision to enter with the constant air charge, whenever the governor permits any admission at all, as is the case with the Crossley. Where throttle controls are used, no recognition is found of the importance of the throttle location, which should be before the vaporizer, so the latter can be under the least possible pressure to promote vaporization, and not after, as in the Thornycroft and Koerting, because here the mixture pressure never falls below atmosphere in the heater and vaporization is not helped.

Any bad adjustment of the proportions in the direction of excess fuel, or the use of bad lubricating oil or an excess of it, will produce smoke and cause exhaust deposits of carbon on the outside of these vaporizing passages. This layer is highly resistant to heat transmission, and as such a layer accumulates the heat received by the mixture becomes less, until a point is reached where vaporization is incomplete and unsatisfactory. At this time the exhaust side of these vaporizer heater walls must be cleaned, but it needs but a glance at those illustrated to see that this is difficult if not impossible in some cases. It should be noted here that if the walls are hot enough, such exhaust smoke carbon will not deposit. This is true, for example, with the externally-ribbed form of the Capitaine, Fig. 14, and the tortuous passages of the Crossley or Fielding and Platt, Figs. 8 and 9. Internal temperature disturbances, such as overheating on the vaporizing side, especially when the oil is vaporizing in a more or less isolated pocket or corner out of and not vigorously scrubbed by the main stream, will likewise cause deposits first of tar and later of

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carbon, which also must be cleaned out. The remedy is first to avoid such constructions and to provide heat control means in the design, and second to make the vaporizing surfaces themselves cleanable, which is not possible in most of the forms illustrated, though all of them have been or are now in successful commercial use for other classes of work than automobiles, where skilled care cannot be expected, as in stationary or even marine installations.

In every case the mixture, consisting of all the fuel, and all or a part of the air, is supplied to these vaporizers in what has been called the rainy state, and as a result there is an early tendency for this to locate on the walls, which tendency is promoted by changes of direction of flow. As already pointed out, a wall film thus formed will tend to concentrate on the low-velocity side of the passage, or drop by gravity or inertia into a pocket, if there is one, instead of distributing over it uniformly. An increased heating surface or size of vaporizer will then be required, and a higher temperature will be necessary to produce dryness just in proportion as these effects are different in one construction as compared with another.

EXHAUST HEAT GOVERNS VAPORIZING CONDITIONS

At low loads the available exhaust heat is not only less in quantity than at full load, but in throttle-controlled engines it is at a much lower temperature normally. For this reason the vaporizing conditions do not remain the same in these vaporizers as the load changes, the tendency being toward deficiency at light load or excess heating of mixture at full load. Low load vaporizing deficiency is corrected in many of the stationary oil engines by using an oil lamp or torch, and even in extreme cases by relying on it entirely, as for example, in the Crossley and Capitaine forms. Of course, this is not admissible in automobile engines, which must rely entirely on exhaust heat for operation, though there can be no objection to a torch or burner for initial starting heat. Exhaust by-passes are provided in the Brouhot and Thornycroft forms to be operated by hand, which also is not permissible in automobiles.

Where exhaust heat is relied on, the vaporizer is large and there is clear evidence of an effort to provide extended heating surface. Small size is secured only where intense oil-flame heating is used, as in the Capitaine. Large heating surface would be necessary if only a small and fixed amount of heat were available for transmission through or mixture absorption from each square foot, and this is the prevailing idea of these older designers. It conforms to the old ideas of steam boiler practice, now abandoned in the light of better knowledge of the laws of heating transmission, which recognizes that the old heat "soaking" theory of constant rate is wrong, substituting for it the modern high velocity "scrubbing" theory, according to which the heating rate is not constant, but increases with velocity.

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HEAT TRANSMISSION AND LOSS OF PRESSURE IN VAPORIZERS

Even if a form of heated vaporizer suitable from the vaporizing standpoint be selected, the size, a matter of great importance, will depend on the degree to which that form conforms to the requirements of the heat transmission laws and at the same time to the laws of loss of pressure through the heater. Loss of pressure means reduced density of charge and power output, and from this standpoint any passage that is excessively long, tortuous or too much baffled is objectionable, especially if the full charge of air must pass through it, as it should for best low temperature vaporizing. Heat transmission rates in British thermal units per hour per square foot per degree mean temperature difference are by no means constant when heat is transmitted from one gas to another through metal plates, and the mean temperature difference itself is subject to even more variation with conditions of heater design and location, for a given engine. These two factors are prime variables in the size of a given heater, and the conditions that result in the maximum values for each and hence the minimum value for the heating surface must be clearly understood and recognized in selecting heater forms, their locations and connections.

Two conditions are to be observed in securing the highest value of the mean temperature difference; first, the condition on the mixture side, and, second, that on the exhaust side. On the mixture side the temperature must be kept as low as possible, and this is accomplished by using the full amount of mixed air in the vaporizer so as to secure dryness at the lowest temperature. On the exhaust side the gases are hottest at the exhaust-valve-port outlet, so that the heater should be as close to this point as possible. It is well known that visible flame, especially the red or yellow kind rather than the blue, has a heat-radiating power, and that radiant heat will pass from such a glowing point through intervening gases into metal walls practically without resistance, and at rates enormously higher than it can pass from a hot gas current through the same walls to a cooler gas current.

The exhaust gases from these engines have radiating power, and radiant heat can be obtained from them by any walls placed in direct line. For this reason not only should the heater be close to the exhaust ports, but it should also lie so directly in front of them as to receive all the radiant heat in these gases before it is intercepted by any header or pipe connection. A heater conforming to these best conditions would, for a multi-cylinder engine, be of elongated form and be placed directly across the exhaust outlets within the exhaust header. If not of this form, it should conform as nearly as possible to the conditions of closeness, although if the radiant heat be lost, more surface must be provided.

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HIGH GAS VELOCITIES AND LEAST LOSS OF PRESSURE

The unit heat transmission rate from hot gases to cooler mixtures follows the law first suggested by Jordan and now established and known as the mass flow law. According to this, whenever the controlling resistance to heat flow lies on the gas side of a wall, the heat that will pass or be taken up by the gas is directly proportional to the weight of gas passing per minute. Therefore, as the amount of heat is the product of the specific heat, the temperature rise and the weight, it follows that in such a case the final temperature will remain constant, no matter how the flow may vary. According to this

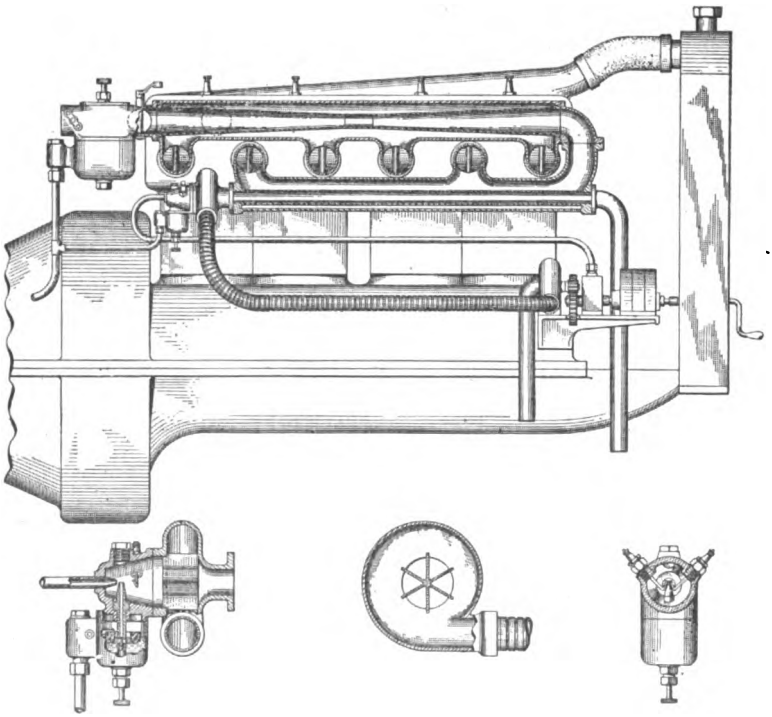

FIG. 15—THE GOOD VENTURI MIXTURE VAPORIZER AND STARTING BURNER

the mixture should at all times be given a high velocity if the high rate heat absorption is to be promoted, by using small passages for the mixture. This, of course, runs counter to the requirement for least drop in pressure in ordinary passages, because high velocity produces pressure loss by friction. The best result is therefore to be obtained by a judicious balance between high mixture velocity on the one hand and excessive loss of pressure on the other, so that the heater size can be a minimum, consistent with least loss of engine power.

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In any case the form of passage that permits of the use of the greatest velocities with least loss of pressure is most suitable, that is, the passage should be as frictionless as possible and no abrupt bends or eddy currents are permissible. Increasing mixture velocities will not necessarily increase the rate of heat transmission, nor will the final temperature be constant unless, according to the mass flow law, the controlling resistance be on the mixture side of the heater walls. Should it be on the other, or exhaust side, the rate of heat transmission will depend on the rate with which the tube metal can get the heat. This also follows the mass flow law with one exception. The exception element is the radiant exhaust gas heat that enters the tube at a rate so much higher as to tend to keep the thermal resistance on the mixture side. Heater location directly in line with exhaust valve discharges therefore tends to establish not only the high rate of heat transmission desired for small heater size, but also to keep the thermal resistance on the mixture side of the wall, which in turn is the condition for automatic constancy of mixture temperature.

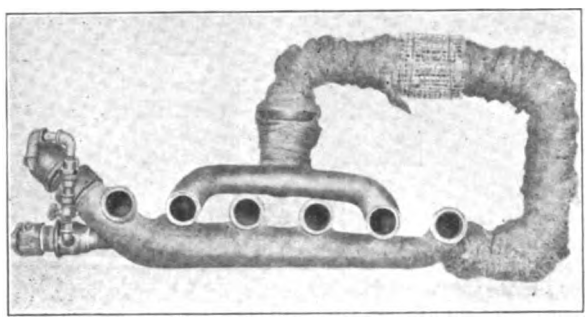

FIG. 16—EXPERIMENTAL GOOD VAPORIZER WITH SIGHT GLASS

To still further promote the same end, the exhaust gases must move past the heater walls with the maximum possible velocity. This is promoted by so locating the heater walls as to make the exhaust gases impinge on it with their high exit velocity, and by keeping the size of the exhaust passage along the heater walls as small as is consistent with non-development of excessive back pressure. By making the heater the restricted exhaust passage, instead of the muffler, the vaporizer will itself act as a muffler, and produce the desired heat transmission conditions with no more back-pressure than is now normal. The heat absorbed by the mixture has been taken from the exhaust, reducing the volume of the latter correspondingly, and hence the back-pressure beyond the heater in exhaust pipes of normal size. It must not be understood by this that mufflers can be eliminated necessarily, but rather that they can be reduced in size or resistance, in proportion as the heater shares the duty.

With due regard to these conditions for high-rate heat transmis-

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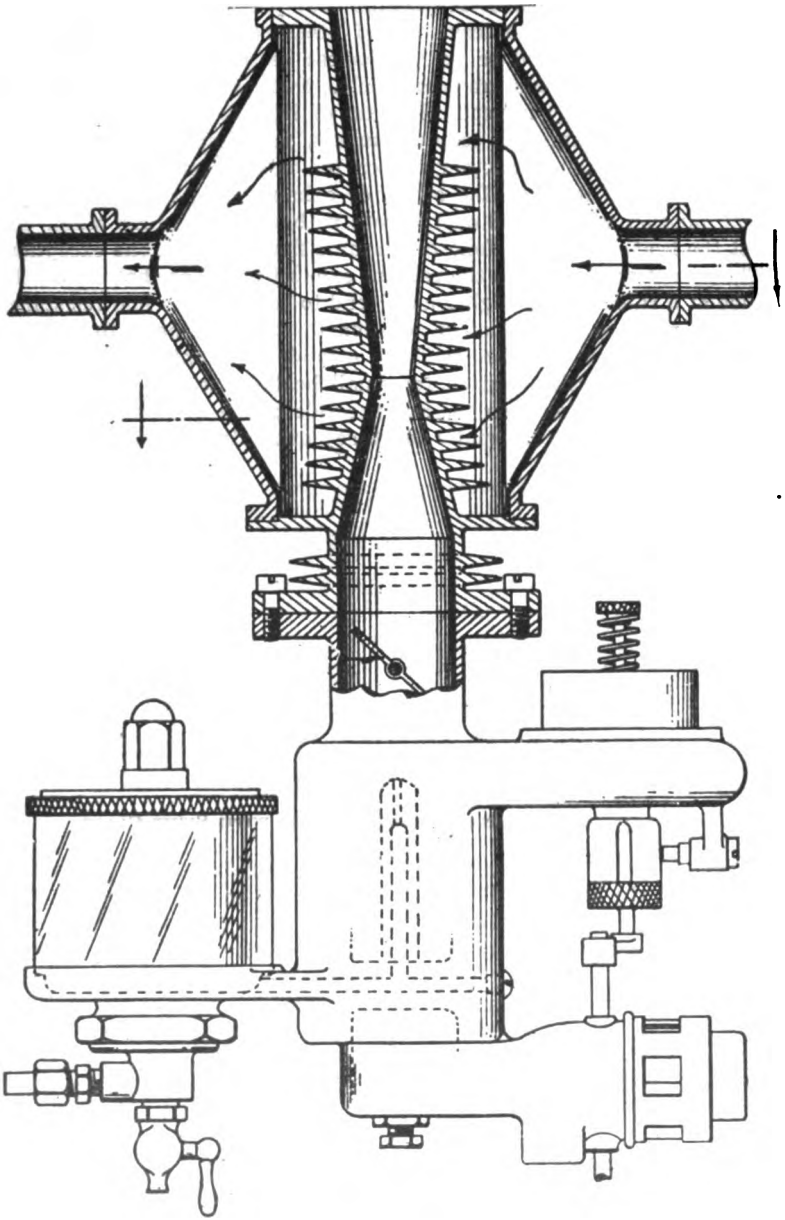

FIG. 17—GOOD VENTURI VAPORIZER WITH CARBURETER AND THROTTLE

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sion, the heater will be small and the mixture temperature as constant as they are perfectly executed. The result will be most nearly perfect when both mixture velocity and exhaust velocity are high, the latter higher than the former; when the exhaust gases impinge most directly and most vigorously on the heater walls; and when these walls receive the maximum radiant heat from the first escaping luminous exhaust gases.

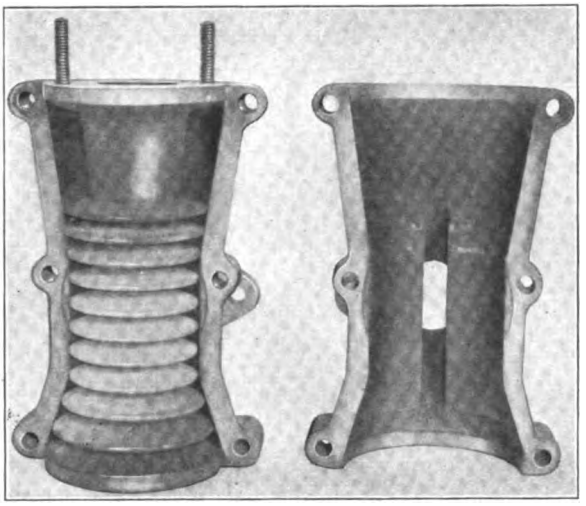

FIG. 18—GOOD RIBBED VENTURI WITH EXHAUST CASING OPEN

The form of the exhaust valve ports can contribute to, or oppose these results; the former when the gases are allowed to come straight out through the shortest possible passage; the latter when, as in block castings, the gases must make a double turn in a water-cooled pocket, which cuts off all the radiant heat and a good deal of the hot gas heat. Similarly, on the intake, short straight inlet ports will chill the warm mixture least; crooked long block-casting pockets, cooled by water, will chill the mixture most. There is little justification for careful design of heaters if inlet valve pockets are left to rob the mixture of its heat, obtained at so much trouble in the heater. Of course, in any case an operative condition is obtainable, but is more easily obtained with smaller and lighter equipment if the engine connections are favorable.

VAPORIZER EMBODYING CORRECT PRINCIPLES

It is possible to design heaters and connections in a great variety of forms, and still conform to the principles developed as desirable and good. No doubt many do so conform, but to make the situation clear and convincing, one form, recently developed by John Good

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of Brooklyn, N. Y., will be described, because this seems peculiarly adapted to the purpose, is most familiar to the writer and has reached the stage of having had successful road trials on a Ford car, after long and systematic study on the test block in competition with many others of great variety. This is presented as one good concrete example of the principles discussed, and entirely without prejudice to others known to the author.

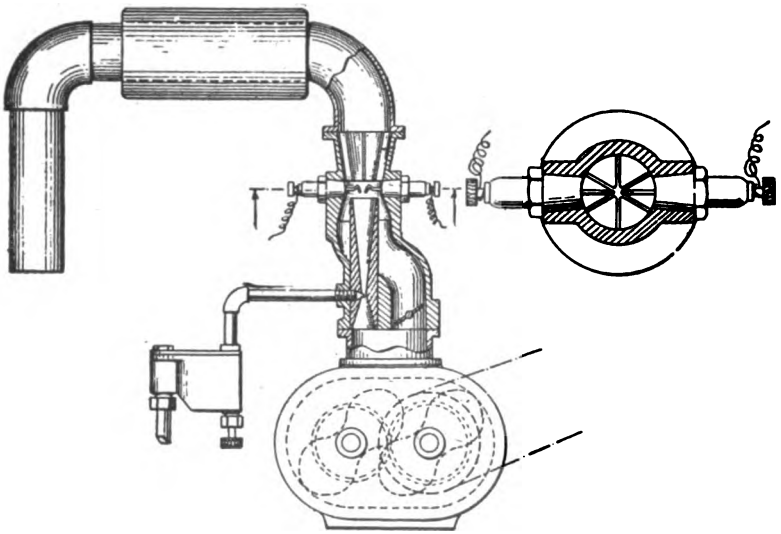

FIG. 19—GOOD SINGLE BLOWER-TYPE STARTING BURNER WITH VENTURI SPRAYER

The Good vaporizer is in the form of a Venturi tube of narrow angle, made of thin steel tubing, which weighs little, is strong, and cleanable on both sides. Its length is made to conform to the exhaust header in which it is mounted, so as to be close to and preferably directly in front of the exhausts. This form of passage, by the well-known laws of such a tube, produces a high velocity at the throat with a correspondingly considerable pressure depression, and with the least possible overall loss of pressure between the ends. The high critical velocity promotes good heat transmission conditions; the throat pressure depression promotes quick vaporization by reducing the total mixture pressure and the partial vapor pressure; and pressure recovery is produced by the enlarging cone, in which velocity is converted back into pressure with practically no loss.

In a series of Venturi tubes, recently tested by Prof. E. D. Thurston, Jr., of Columbia University, with throats ranging in diameter from ¼ to 1 in., and having angles of 11 deg., the overall loss in pressure for the ¼-in. diameter throat was 3 in. of water at 10,200 ft. per min. throat velocity, and 1½ in. of water at 6000 ft.

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per min.; while in the 1-in. diameter throat the loss was about 50 per cent more. This is practically nothing and much higher velocities are quite feasible. The high velocities, characteristic of Venturi throats, are above those analyzed by Osborne Reynolds and characterized by "turbulent flow;" as a result the mixing of the air and vapor is perfect, and far better than can be obtained by baffles, which always introduce friction and loss of pressure.

This Venturi heated vaporizer is shown in place in the exhaust header, Fig. 15, with an ordinary carbureter feeding it with a wet, rainy mixture. The warm dry mixture discharges around the end to the intake header just below, this engine having both valves on one side. The connections were such that it could not be placed in the most favorable position, directly in front of the exhaust outlets. An early experimental Venturi vaporizer, Fig. 16, is shown with two headers equipped for studying dryness by the wire-screened glass

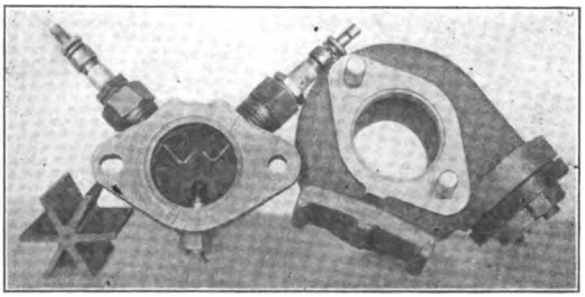

FIG. 20—PARTS OF GOOD DOUBLE-BLOWER ATOMIZER-SPRAY OF STARTING BURNER

sight-tube, and with lagging to save loss of heat in the long bends inserted to get a good position for the sight glass. Another and shorter form, Fig. 17, has external ribs intended to be placed vertically at the end of the exhaust header, the mixture carbureter being below and the throttle-valve located between carbureter and vaporizer to get the full advantage for vaporization of all pressure depressions, due to a partly closed throttle. An external view of this is shown in Fig. 18.

STARTING BURNER FOR KEROSENE ENGINES

No vaporizer for automobile use can of itself be regarded as furnishing a solution of the kerosene problem, because starting heat is necessary, and this must be derived from the burning of kerosene in a suitable burner. The old suggestions and practices of starting on gasoline, or, of using electric heaters, are not satisfactory, the first because it is a nuisance, and the second because of the large storage battery capacity and the high rate of electrical discharge required. A good semi-automatic starting burner also has been developed by Mr. Good, and has proved to be thoroughly reliable. This burner,

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together with an independent starting heater, is shown in Fig. 15. The burner is located at the end of the intake header, and its flame passes along the inside of a small light steel tube, inside and close to the bottom of the header. The burner flame raises this interior tube to a red heat in about fifteen seconds, the products of combustion passing out the end and downward to a point below the engine. No flame ever appears at the outlet. The good quality of combustion and the shortness and intensity of the flame are indicated by an analysis of the products, which showed only two per cent of free oxygen and zero carbon monoxide.

The construction of the burner is shown by the lower right-hand section where the vertical oil nozzle, fed from a float-chamber, is crossed horizontally by a jet of air at about one pound pressure, pro-

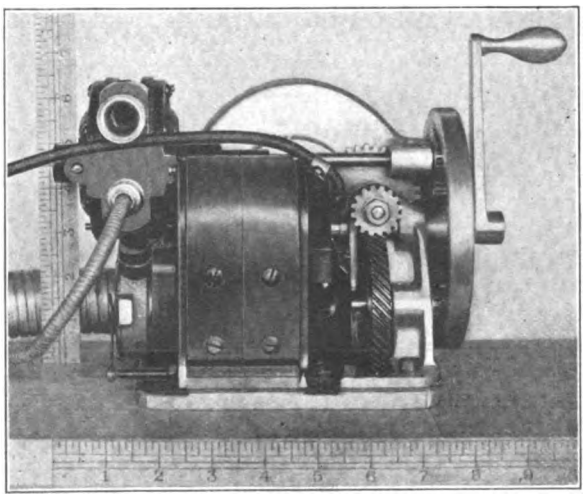

Fig. 21—Assembly of Blower, Magneto and Fan for Burner

duced by the smallest size of Wing positive blower, shown beside the magneto. The crossing air jet makes a fine spray of oil which moves past the spark points where it is ignited automatically by the magneto geared to the Wing blower. The spark points are arranged as shown in the lower right-hand section (Fig. 15) with the terminals bent up and toward each other. This causes the oil drops that collect to fall away to the lower loops by gravity, leaving the tips free to spark; before this was done the ignition was bad, but afterward it never failed in over a year. The spray flame escapes through an enlarged air chamber, supplying air around the spray to support the combustion, which becomes vigorous only beyond the radial guide vanes at the exit from the burner or at the entrance to the heating tube, tightly fastened to it. The air supporting combustion is supplied

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through a flexible metallic tube from a small vacuum cleaner fan, geared to the Wing blower and the magneto. One turn of the shaft automatically starts the burner and it continues to burn as long as the motion is applied, stopping automatically when the motion stops.

In another form of this burner, one larger positive blower is made to take the place of the fan and small blower; this is shown in Fig. 19. Here the main air is divided, one part passing through a small Venturi tube, meeting oil at the throat where it is sprayed by the high velocity. The main air stream is by-passed and led around the oil jet in an annular layer surrounding the oil spray to support combustion. The spark-plug terminals are bent toward the high velocity jet so the collected oil drops will be driven back and away, independently of gravity.

The parts of one of these burners of the first form are shown externally in Fig. 20. As assembled, the magneto, Wing blower and fan, geared together for operation by a hand-crank are shown in Fig. 21, in which is an inch scale, showing that its overall length is about 7 in. and its height is 6 in. A complete burner outfit is shown in Fig. 22, where the burner is attached to a heating jacket welded around a Ford inlet header, so that the starting heat is applied to it externally, as compared with the internal arrangement of Fig. 15.

OPERATION OF STARTING BURNER

This equipment, applied to an engine on the test block or to a car on the road, works with satisfaction so far as operation is concerned. There is no more trouble than with gasoline except a slight starting lag, measured in seconds, owing to the necessity for letting the starting burner develop the necessary starting heat. The engine idles and accelerates just as well as with gasoline; there is no smoke or carbon accumulation, and the operator would never know he was using kerosene, and is more independent of cold weather than with gasoline as now used. With hand-cranking the burner gear can be rotated by the starting handle a few times before engaging the engine shaft; then by pressing in the handle to pick up the engine, it will start promptly. It is desirable, although not necessary, that the burner be continued in operation a few minutes after the engine is started, and until the interior and the exhaust connections warm up. This will prevent entirely the temporary accumulation of kerosene in the cylinder while it is cold and obviate completely smoke and lubrication interference. It is not absolutely necessary, as such accumulations are slight and disappear quickly as the engine operates, and in no case do they prevent its running.

With a motor-starting gear the whole engine and burner can be started simultaneously, the first oil passing through without igniting and some, of course, accumulating until the burner tube heats up, which it does very quickly, and then the engine picks up and evaporates the oil accumulation. The whole time required and results are substantially the same as are now common with heavy gasoline

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in cold weather. Even this condition can, however, be eliminated entirely by a little electric fan motor on the burner gear, started by a button a moment before the main engine motor starting button is operated. With this arrangement no kerosene can accumulate, the engine should pick up instantly and the operator can keep the burner in operation as long as conditions warrant.

AUTHOR'S CONCLUSION

It is hoped that by this presentation of principles on which good and satisfactory kerosene equipment can be designed, illustrated by one concrete example, the movement toward the production and use of kerosene automobiles will be undertaken immediately on a scale sufficiently large to affect the price of fuel within the next year. Commercial results can be obtained most quickly and with the least development and experimental expense, by the adoption of the sort of equipment described, but such adoption is not an absolute necessity. A hundred different sorts of equipment could undoubtedly be

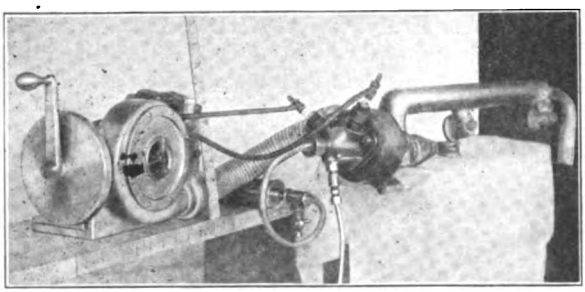

FIG. 22—WORKING RELATION OF GOOD STARTING BURNER AND AUXILIARIES

designed all conforming to correct principles, but only with the expenditure of more experimental and development funds and with corresponding delay in time. Finally, it is equally possible to proceed along quite different lines from those laid down as the most desirable and immediately available, and still get good results. The public in general and the automobile and the oil industries must have something that will use kerosene and be so acceptable as to secure use and adoption on a big scale. In the long run it matters little just what principles are used in the equipment as long as it works right, but it is somewhat comforting to know at this time that something is immediately available for designers to adopt and use.

DISCUSSION

P. S. TICE:—Dr. Lucke's paper is of exceptional interest because of the manner in which he has summarized the general conditions that must be satisfied in handling kerosene. I wish to supplement the paper with some data showing the conditions that must be taken

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into account and with which we have to work in the application of kerosene to a modern motor car.

If we assume a car to be running on a smooth level road, in high gear, at various speeds from idling to maximum, the curve of fuel consumption is of the form at *A;* running under open throttle over the same speed range, the characteristic curve of fuel consumption is as at *B*, Fig. 23. Under these two sets of conditions the corresponding curves of exhaust temperature are as at *A* and *B*, Fig. 24. Knowing the ratio of air to fuel in the mixture, the weight of fuel burned, and the temperature of the resulting exhaust gases, one can determine the available heat in, or heat content of, the exhaust under the two sets of running conditions of Figs. 23 and 24. Curve *A*, Fig. 25, shows the heat content of the exhaust with the car operated under part throttle; curve *B* the heat content of the exhaust under open throttle.

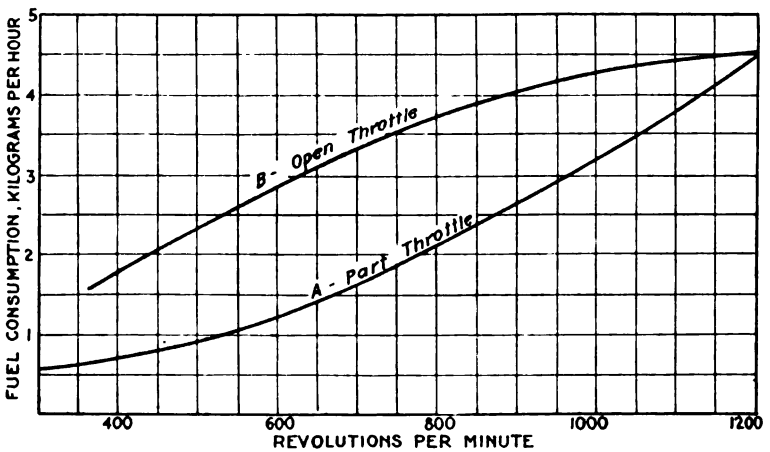

FIG. 23—CHARACTERISTIC CURVES OF FUEL CONSUMPTION

From the form and position of curve *A*, Fig. 25, it is apparent that the greatest difficulty in operation can be expected at the lower car speeds under comparatively close throttle. The heat content of the exhaust is then relatively small as a result of its lesser mass and comparatively low temperature. Curve *C*, Fig. 25, of heat required to evaporate the fuel used by the engine in level-road operation shows that this is true; ample heat is available at a sufficient potential however if the arrangements for its utilization are suitable. With anything other than the nearly closed throttle positions, we have more leeway, as the curves show. Curves *A* and *C*, Fig. 25, provide a graphic argument for the location of the vaporizing chamber close to the exhaust ports of the engine, as advocated by Dr. Lucke. Ordinarily, if only 8 to 10 in. of pipe intervene between the exhaust ports and the vaporizer, it will be impossible to idle or run slowly for a

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reasonable length of time without smoking and more or less disturbance of operation.

In the system described by Dr. Lucke, the mixture temperatures will be too high for suitable operation in a car unless great sacrifices are made in the compression pressure. Prof. W. Morgan states in his discussion* of Dr. Watson's last paper on fuels that at a compression temperature of 373 to 378 deg. C. (704 to 712 deg. F.) a mixture of gasoline vapor and air will automatically ignite; and that kerosene mixtures begin to burn at 309 to 313 deg. C. (588 to 595 deg. F.). According to these figures, with which my experience is in entire agreement, the terminal compression temperature must be lower with kerosene than with gasoline mixtures. On the other

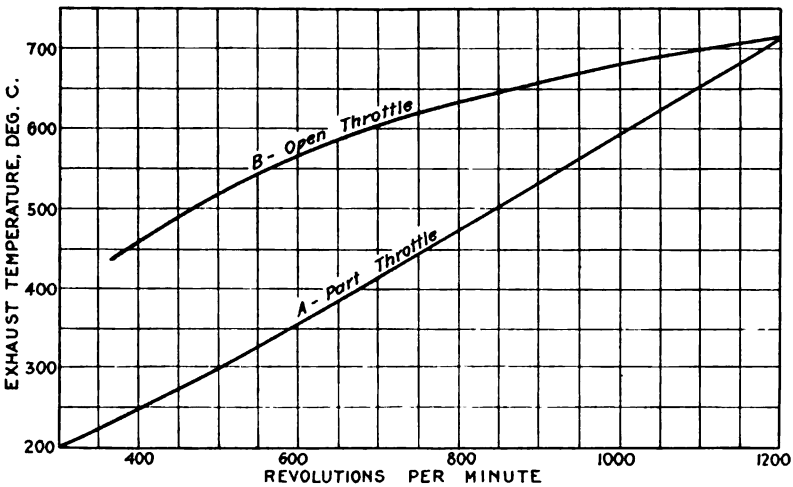

FIG. 24—CHARACTERISTIC CURVES OF EXHAUST TEMPERATURES

hand, in order to use kerosene at all we must make certain that it is fully vaporized. Because of loss in output and unqualified and uncontrollable pre-ignition at anything over half-load, it is a mistake to exhaust-jacket the intake pipe; the manner of heat transfer to the fuel under such conditions is not good because the whole mixture is heated. I have found that with systems of this sort with vaporization secured under close-throttle operation, the mixture temperature at the intake ports will run up between 177 and 204 deg. C. (350 and 400 deg. F.). Under open throttle that is so high that the power falls off, and the fuel consumption is extraordinarily high—as much as 3 lb. per hp. hr.

With a portion of the intake passage jacketed in such manner

*Proceedings (1914-1915) Institution of Automobile Engineers, p. 99.

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as to give fair low-speed operation, it was found upon starting an open-throttle run at the speed of maximum torque of the engine that the torque reading was 63 lb., which is a normal value for the engine used. After a few minutes' running, when normal open-throttle temperature had been reached, the torque dropped to 41 lb. and remained there. This represents both the volumetric loss following too great heating of the mixture and whatever other losses are introduced as a result of the susceptibility of the mixture to too early and violent combustion.

EVAPORATION OF FUEL

There are several expedients for obviating this condition; the most promising is to develop a fog-mixture system, to borrow from

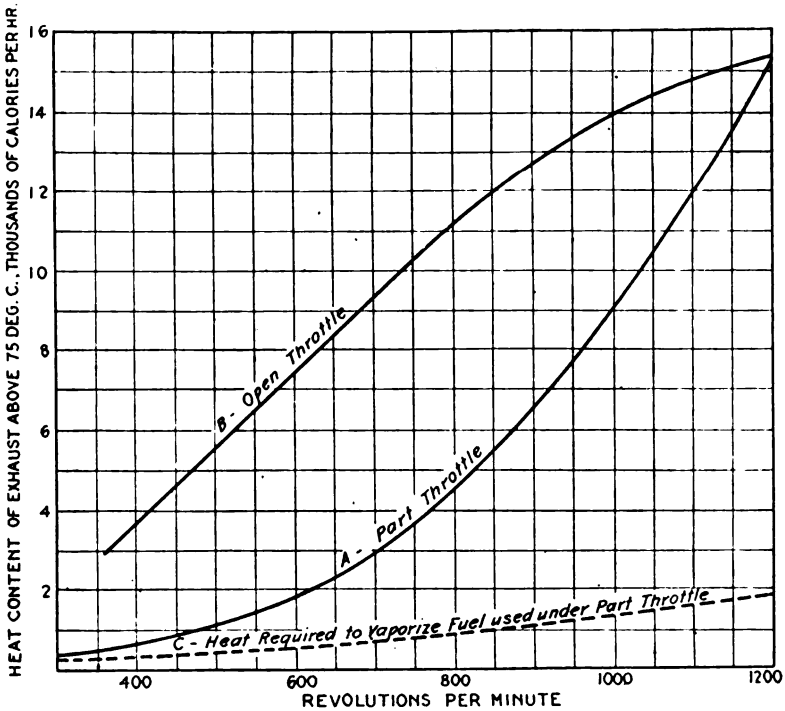

FIG. 25—CHARACTERISTIC CURVES OF EXHAUST HEAT-CONTENT

Dr. Lucke's terminology. First evaporate the fuel, and then mix its vapor with the air. I have developed a method in which is incorporated the essentials for the highest possible rate of evaporation of the fuel in the presence of the whole of the air with which it is mixed, and in which unduly high mixture temperatures are obviated.

The method in question involves a rather peculiar carbureter construction, Fig. 26. The throttle is in the air-intake of the carbureter, and there is no passage obstruction between the nozzle and the

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cylinder ports. The dome directly over the carbureter outlet is located within the exhaust manifold in such a position that the gases from one or more of the exhaust ports impinge directly upon it. Because of the location of the throttle in this carbureter, and of the fact that an "air-jet," so-called, is in the center of the nozzle, the spraying with normally-used throttle openings is extremely energetic. The curve, Fig. 27, shows how the spraying head on the nozzle air-jet varies under level-road running conditions, as noted for curve *A* in the preceding figures.

The kinetic energy imparted to the fuel particles by the air-jet is sufficient to cause them to leave the air-stream at the branching of the intake pipe, and continue into the exhaust-heated dome or vapor-

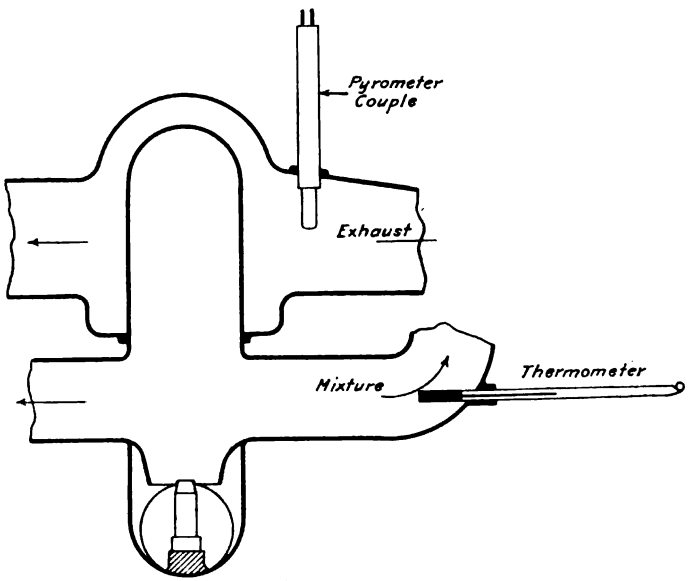

FIG. 26—TICE CARBURETER CONSTRUCTION WITH VAPORIZING CHAMBER

izing chamber. This is provided in practice with longitudinal ribbing on the interior wall and with circumferential ribbing on the exterior wall. The fuel particles leave the nozzle in the form of a narrow-angle cone surrounded by the air-stream. At the branching of the manifold, the air makes the turn, with perhaps a slight swing toward the mouth of the vaporizer, while the fuel particles pass directly through it and are distributed over the vaporizer surface. Here evaporation is practically instantaneous, and the vapor passes out of the vaporizer and mixes with the air-stream. Because of the general structure and method of operating the nozzle, the fuel is so finely divided that, with a suitably proportioned vaporizer, the interior surface never becomes wet, and there is no evidence of the spheroidal

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condition for the same reason. These statements are the result of direct observation through windows placed in the experimental apparatus.

As the kerosene vapor passes from the vaporizer and reenters the manifold proper, there are two clearly defined lines of demarkation, beyond which, entering the manifold branches, the mixture looks foggy and fills the passage. As Dr. Lucke aptly says, it has a tobacco-smoke appearance. From this point on we have a mixture of condensed vapor (otherwise we could not see it, of course) and air. At no time in the process has there been direct heating of the air

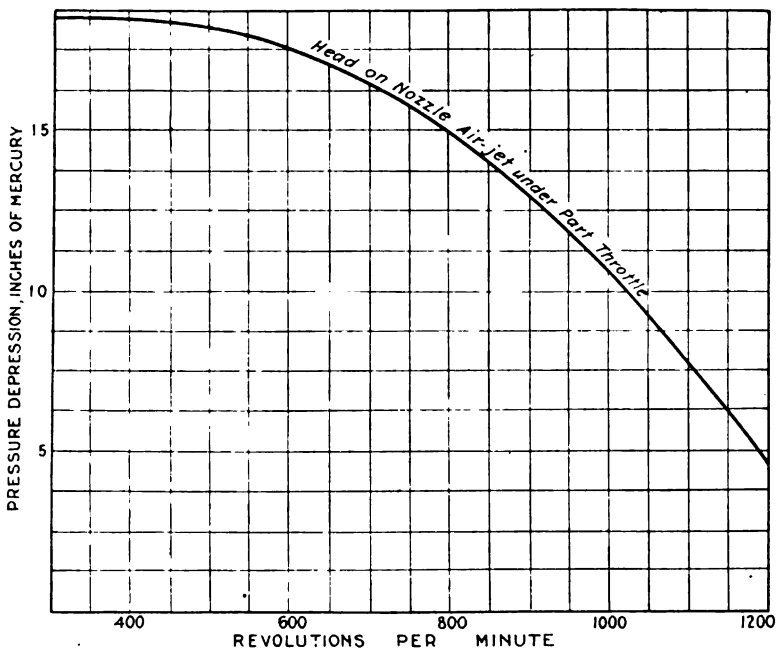

FIG. 27—EFFECTIVE SPRAYING HEAD IN TICE CARBURETER UNDER PART THROTTLE

forming the mixture, except what must occur through conduction of heat through the metal of the manifold. Otherwise the whole of the heat appearing in the air of the mixture as it approaches the valve ports is represented by heat given up by the fuel vapor upon its condensation as it enters the air-stream. With a thermometer located as shown in Fig. 26, the mixture temperature under conditions of level-road running is represented by curve *A*, Fig. 28. The mean value of this curve can be made to coincide with the mixture temperature that gives the greatest economy. In the case of kerosene this temperature of greatest economy is nearly 77 deg. C. (170 deg. F.).

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By increasing the capacity of the vaporizer in this particular system the mixture temperature will be raised through superheating of the vapor and its consequent giving up to the air of a greater amount of heat. Similarly, decreasing the capacity of the vaporizer will cause a lowering of the mixture temperature.

HERBERT CHASE:—Dr. Lucke says (page 129) "Such mixtures (referring to the gases being taken into the engine) are hot, necessarily as hot as the fuel vapor pressure requires, and therefore the compression must be reduced, reducing horsepower and efficiency correspondingly but not to a serious degree." This, I take it, refers to the dry mixtures. Later the author states that a temperature of 121 deg. C. (250 deg. F.) is necessary to get this dry mixture into the inlet manifold. Under such conditions the volumetric efficiency,

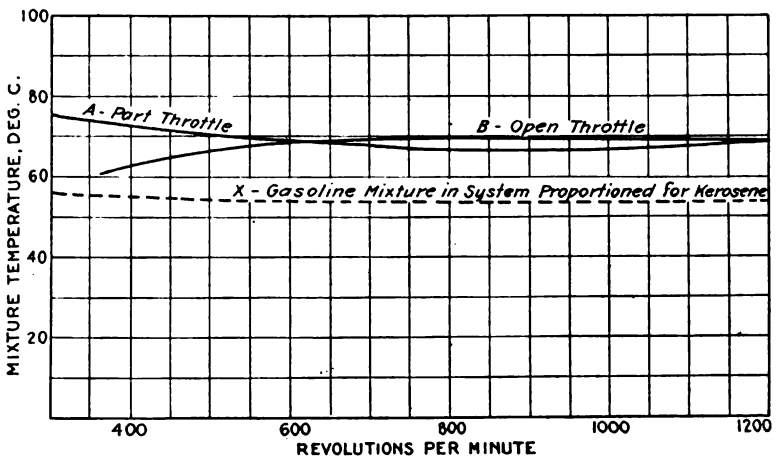

FIG. 28—KEROSENE MIXTURE TEMPERATURES WITH TICE VAPORIZING SYSTEM

and of course the power, will certainly be decreased. This is a step in the wrong direction if we are to continue our efforts to get maximum power from a cylinder of a given size.

I am of the opinion that the best solution of this problem is to design an engine, which from the start is suited to the use of heavy fuel, rather than to attempt to use any device on existing types of engines that will render them less efficient or less powerful for a given size and weight.

P. S. TICE:—As a result of my experience no redesigning of the engine proper seems to be necessary. My work has been done largely with an engine having a detachable head; at the outset a number of heads having different clearance volumes were secured, so that by them the compression pressure could be varied from around 38 to over 75 lb. gage. The head giving a compression of about 65 lb.

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was finally settled upon as being fully suitable with a fuel system wherein the mixture temperature is maintained at a reasonable value. Above 65 lb. misadjustment or misfitting in the system can give rise to trouble; while if the compression is below that value the volumetric efficiency is unnecessarily reduced, and one cannot get the full output from the engine.

MIXTURE TEMPERATURES

CHARLES GUERNSEY:—As I understand Mr. Tice's diagram the temperature he has shown for the fog mixture, about 77 deg. C. (170 deg. F.), is under closed throttle. He pointed out that there is a great variation in the heat of the exhaust between close and open throttle. I would like to know the temperature in the mixture with open throttle.

P. S. TICE:—Curve *A*, Fig. 28, while it has been called mixture temperature under part throttle, actually represents the full-throttle condition at the highest speed shown, as do all the *A* curves in the diagrams given. Curve *B*, Fig. 28, shows mixture temperatures under open throttle at the lower speeds. This curve is practically horizontal over the upper two-thirds of the speed range and drops slightly at the lower speeds. But this drop is accounted for by the difference in the mixture proportions at the lower speeds under full throttle as compared with like speeds under close throttle.

The explanation of this fact is that there is no direct heating of the air other than that due to the heat conducted back through the metal of the members; the chief source of the air-heating is that heat taken away from, or given up by, the vapor as it condenses upon mingling with the air in the system. The temperature of the air, before the vapor mixes with it, is but little above the temperature of the atmosphere, while the temperature of the vapor, before mixing with the air and as it issues from the vaporizing chamber, is extremely high. The lowest vapor temperature of which I have record, with a vaporizer of sufficient capacity to give complete evaporation, is 129 deg. C. (264 deg. F.).

Normally the vapor temperature cannot be higher than that value at which the heaviest parts of the fuel mixture evaporate, because of the continuous presence of the fuel in the liquid state, represented by the spray moving in opposition to the direction of flow of the vapor. Therefore any rise in the terminal mixture temperature represents an enrichment of the mixture, in its proportions of fuel to air. Likewise a lowering of the mixture temperature follows any reduction in the relative amount of fuel, since there can be no superheating of the vapor in the presence of the liquid spray. The amount of heat given up to the air by the vapor and therefore the terminal mixture temperature are naturally directly proportional to the relative amount of fuel in the mixture.

Naturally, we do not maintain these proportions constant, and it is undesirable to do so over the whole range of operation. Because

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of residual gases left in the clearance spaces under close throttle, and under open throttle at the highest speeds, slightly richer mixtures are desirable under these conditions than under open throttle at the lower and intermediate speeds. These facts account for the forms of the curves of mixture temperature. The richer the mixture the higher the mixture temperature, and vice versa; with richer mixtures more fuel is evaporated and as a consequence more heat is given up to the air because there is more vapor to condense from a given temperature.

GASOLINE IN CARBURETING DEVICE

HENRY SOUTHER:—What happens if gasoline is used in that same carbureter device?

P. S. TICE:—Substituting gasoline for kerosene in such a system, the curve of mixture temperature (*X*, Fig. 28) averages some 15 to 17 deg. C. (28 to 30 deg. F.) lower than that for kerosene. The explanation undoubtedly is that the gasoline evaporates at a lower temperature and has a lower value for the total heat of vaporization; thus if saturation is maintained in the vapor there is less heat to be given up to the air by the vapor. A system proportioned for kerosene has a somewhat greater capacity than is essential for gasoline, and will maintain the gasoline-mixture temperatures at higher values than are necessary.

The figures of Professor Morgan are extremely interesting, and I think explain the bumping and pounding ordinarily experienced with kerosene in the applications that have been made. If the fuel is handled to secure complete evaporation, there is no bumping if the mixture temperatures are maintained within such bounds that the terminal compression temperatures do not exceed the value at which occurs what Professor Morgan calls incipient combustion. His figures were obtained by running the mixtures in question through a tube heated in a bath of molten metal and gradually raising the temperature of the bath until combustion was started. We must grant that he worked to close limits, and further assume the accuracy of his work. Temperatures of 189 to 192 deg. C. (373 to 378 deg. F.) for gasoline and 154 to 156 deg. C. (309 to 313 deg. F.) for kerosene show close readings for such work.

HENRY S. OTTO:—Is the mixture dome mentioned by Mr. Tice preheated in starting?

P. S. TICE:—In my use of the system I started on gasoline. On the driving side of the dash was mounted a 1-qt. tank fitted with a needle-valve discharging into a ⅛-in. o.d. tube, which was led to the carbureter at a point inside the throttle. The fitting by means of which this pipe was attached to the carbureter had a small admission hole for the gasoline and a small transverse hole to admit air. Starting was promptly accomplished by opening the needle-valve at the tank and kicking the starter, with the carbureter throttle nearly closed. There was no difficulty about the starting, and it required

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no particular regulation. In winter weather, during which my final experimental work on this device was done, I had to run from 4 to 5 min., from time to time cutting down the supply of gasoline at the needle-valve. At all times, from the time of initial start, the normal amount of kerosene was being aspirated from the carbureter; there was smoking at the start and for from 2 to 3 min. thereafter, and then things began to clean up. After from 4 to 5 min. running with the auxiliary, the gasoline is completely shut off, and the device performs in its normal manner. In summer weather with the engine cold the start is made in about half that time. With suitable vaporizing-chamber proportions, even in winter weather, and without an overcoat on the radiator and hood, one can start directly on kerosene in about an hour after the engine has been stopped.

W. P. DEPPÉ:—Dr. Lucke has gone into this subject more deeply than most men not in the active work of developing gas generators to use either gasolines or kerosenes in standard engines; yet he is perhaps not fully aware that gas generators do run on either gasoline or kerosene in many types of engines, with heat sufficient not only to vaporize but actually to produce superheated gas in the cylinders. These generators do not depend on the heat of compression or flame spread to assist vaporization.

The introduction of kerosene engines and also of gas generators not able to handle any grade of gasoline will not solve the problem finally. What the oil men and leading motor-car engineers, so far as we learn their views, wish is something to handle either fuel at the present time.

USE OF LOW-GRADE GASOLINES

I think that General Manager Reeves of the National Automobile Chamber of Commerce hit the nail squarely on the head when he said that in spite of all the wonderful development in the automobile industry the crying necessity is a carbureter that will use 54-deg. Baumé gasoline. Such gasoline means in fact the present gasolines with kerosene cracked into so-called gasolines (possible commercially through the Burton, the Rittman or any other cracking system) with 40 per cent of ordinary kerosene added. It is unfortunate that when cracked gasolines are converted from the liquid by any known method, the resulting gases and vapors act in the engine cylinders just as if they were the vapors and gases of the true kerosenes from which such gasolines were cracked.

Cracking methods make it possible for the oil refiner to sell his kerosenes at gasoline prices. If gas generators able to handle either kerosenes or gasolines are installed generally on engines, the public can for a while at least buy present grades of gasolines at slightly lower prices. It can also buy 40 or 50 per cent of the total oil fuels it uses at kerosene prices, if the car makers will only see the advantage to them of having their cars equipped so that the individual can

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use good gasoline, cracked gasolines, kerosenes or any mixtures desired.

Perhaps final solution will be the force of economic pressure, compelling oil refiners to make a straight distillation down to 54 deg. Baumé from the crudes, without splitting gasolines. For the bulk of the world's supply for internal-combustion engines, the refiners can then make a limited supply of high-grade gasolines, true naphthas, true benzines, and true kerosenes. Surplus kerosenes from such limited distillation of crudes can readily be dumped into the low-grade gasoline. The low-grade gasolines could be sold for reasonable prices, while the true gasolines, naphthas and benzines should perhaps sell for higher prices than at present.

All gas engines have only about half of the fuel efficiency possible in commercial types. At the rate the oil fuels are decreasing in volatility, another five years will prevent the use of all outstanding motor cars. It will be increasingly apparent that all existing engines will go to the scrap-heap.

VICE-PRESIDENT FOLJAMBE:—Most of this discussion has been on the intake pipe and the carbureter, although other difficulties are encountered. For instance, the bosses or any masses of metal in the cylinder will affect the results. I know of tests in which the engine smoked a great deal. The intake passages were all right. It was thought that certain bosses had enough metal around them to cause cracking of the fuel in the cylinder. By merely squeezing cold water from a sponge on the portion of the engine near the bosses, the exhaust instantly became clear.

AUTHOR'S CLOSURE

DR. C. E. LUCKE:—The discussion indicates not only that interest has been aroused in kerosene, and that people are beginning to realize that previous failures in the use of kerosene do not prove that it cannot be properly used, but also that there is some productive activity along the lines of developing suitable appliances. If a sufficient number of people working under the guidance of correct principles can be kept at work on this problem, not only one solution but as many different ones as will satisfy the industry as a whole, will be assured.

The contribution of Mr. Tice is valuable, inasmuch as it presents the results of some productive work along the lines of developing another one of these appliances; however, it is open to criticism when it departs from the presentation of the Tice apparatus and undertakes to generalize on the experiences of others. It is most difficult in dealing with questions of this sort to interpret correctly a description or a drawing for the purpose of drawing conclusions as to how it will behave. Mr. Tice asserts that the Good system of heating the mixture passing through a Venturi tube by exhaust gases will make the mixture temperature too high unless great sacrifices are made in compression pressure, and that the temperatures

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of the mixture will vary widely. Since actual tests prove that this is not so, argument must give way to facts. The combined effect of reduction of compression and excessive temperature would be a direct reduction of power, and the following figures will show that this is not excessive. A standard truck engine recently gave 24 hp. when operating on the Good system at 1000 r.p.m., whereas on gasoline it normally gives 29 hp.

The mixture-temperature fluctuations might be measured in degrees, but they can also be measured by the effect. The most severe test, perhaps, is that of a long period of idling, followed by a sudden opening of throttle. The fact that this can be accomplished with the Good system without any choking, fouling or the commonly observed clouds of smoke, is a practical proof that there is no difference of temperature. This, together with the approach of the gasoline output for full throttle, is a measure of the other extreme of the maximum-mixture temperature. The fluctuation would be somewhere in the neighborhood of 20 deg. F., depending somewhat upon the port construction of the engine and the size and location of the Venturi tube. The mixture temperature is so constant with this Venturi-tube heater as to warrant the characterization on practical grounds of substantially no change in temperature.

One hears at present a great deal about maximum power. The efforts of most of the kerosene-appliance designers have been defeated because of their insistence on this as a criterion. Any engine operating on kerosene under the conditions that will give maximum power cannot remain clean, but cleanliness as good as that obtained with the gasoline ordinarily used, and even better, is had when the mixture temperature rises to the value necessary for dryness with a corresponding decrease of power. Engineers must therefore in dealing with kerosene choose between a moderate loss of power with indefinitely long clean running on the one hand and maximum power with a dirty engine that must be cleaned at short intervals on the other.

Mr. Deppe's belief that the ultimate solution of the problem requires an apparatus that will handle any grade of fuel appeals most strongly, as does also his suggestion that refiners make a straight run down to 54 deg. Baumé without attempting to make any intermediate grades or to crack the heavier products. This can properly be carried even a step further because any apparatus that can use present-day kerosene as fuel can also properly and effectively handle any mixture and give satisfactory service on a straight run from the stills to include everything from gasoline down to gas-oil. This would seem from every standpoint to be the logical thing to do. The engineers can design and adapt such carbureters and heating systems to handle this straight run of the refiners, who will reduce costs by putting out a single product as engine fuel, including within it both the light and the heavy products.

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PROBLEMS IN HIGH-SPEED ENGINE DESIGN

By A. P. Brush

(Member of the Society)

Abstract

The author outlines in a general way the relation of car performance to modern engine development. He considers particularly weight reduction and torque performance of high-speed engines, giving the undesirable characteristics attending the increased torque range gained by higher speed.

He next discusses the relation of torque to total car weight, to acceleration and to hill-climbing ability and suggests a method of determining the value of a car in terms of its performance ability.

The author holds incorrect those systems in which the amount of lubrication is in proportion to speed only; and in which oil for crankshaft and crankpin bearings must cool as well as lubricate them. He shows a system designed to solve these oiling problems.

Static, running and distortion balance of a rotating mass are defined by the author, who shows how they apply to a large number of types of crankshafts. The paper not only deals with the counterbalancing problem as regards the crankshaft itself, but also with the centrifugal effects of the connecting-rod and piston parts. In conclusion a discussion is given of what are correct counterbalancing masses for service conditions.

Before considering the problems involved in the design of high-speed engines, I shall make the statement that any increase in engine speed is, *in itself*, a detriment and not an advantage, and that the only reason for the present tendency to increase engine speeds is that it is the only known means of achieving certain other greatly desired ends.

From the user's standpoint, these ends, for which we must pay by an increase in engine speed, can be grouped under the single head of quality of performance. For the purpose of engineering analysis, I shall classify them as weight reduction and torque range.

For special purposes, such as for racing cars, weight reduction in proportion to maximum power is, of course, of prime importance, and torque range is of relatively little importance; but for every-day all-around service in the hands of the average user, the torque range of an engine is the most valuable end to which higher engine speed is the accepted means.

It is for this reason that I shall not discuss primarily the problems involved in the building of an exceedingly high-speed and an exceedingly light-weight engine, but rather the problems involved in securing increased torque range by means of increased engine speeds.

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Wider torque range in an engine means, in terms of car performance, sustained drawbar pull throughout a wider range of car speed on top gear.

All other things being equal, that car is most pleasant to drive that on top gear has the highest drawbar pull in proportion to total road weight throughout the greatest range of car speed below, let us say, 60 m.p.h. A 40-hp. two-cylinder double-acting steam engine with a maximum speed of 1500 r.p.m. would have a wider torque range than any internal combustion engine of any type or any speed.

The steam engine would have a torque range of 100 per cent, whereas no automobile internal combustion engine has yet been produced with any sustained zero-speed torque. Compared with the best internal-combustion engine of the same power and installed in a car of equal weight, the steam engine will, because of its 100-per cent torque-range, be much the pleasanter engine to operate, for the simple reason that 100 per cent of torque range in the engine means 100-per cent speed range in the car with approximately equal drawbar pull.

I am using the steam engine merely as a convenient comparison and not advocating it as a substitute for the internal-combustion engine, although I must confess that the steam engine is much superior to the internal-combustion engine. The steam engine has lost its place in our estimation in spite of its own desirable and much desired characteristics, because of the bad company it keeps, namely: the steam boiler and the steam boiler's accomplices.

DIFFICULTIES IN SECURING WIDER TORQUE RANGES

The best we can hope to do with the Otto-cycle internal-combustion engine is to develop it toward the torque-range characteristics of the steam engine. Securing in the internal-combustion engine, by means of higher speeds, a torque range approaching that of the steam engine has a tendency to develop undesirable characteristics that can easily, from the user's standpoint, more than compensate for the advantages gained. Chief among these are (1) complication; (2) noise; (3) high fuel consumption; (4) lack of durability; and (5) vibration.

In general, with smaller cylinder dimensions and consequent possibility of smaller valves and lighter non-uniform-motion parts, higher torque per cubic inch of piston displacement can be secured at higher speeds without a corresponding loss of such torque at lower speeds; and particularly in V types, the smaller dimensions permit the design of engines of any desired power that are at once compact, relatively light in weight and with an exceedingly high torque range.

Obviously these advantages are secured only by multiplication of parts. This naturally tends toward the objectionable characteristics of higher initial cost and higher maintenance cost, since a greater number of parts must be maintained in correct working condition if the advantageous results are to be maintained. I use the expression

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"tends toward the objectionable characteristics" because reduction in size of non-uniform-motion parts has a counter-tendency toward less first cost and greater durability of each of such non-uniform-motion parts. The tendency toward greater durability is due to the natural law that structural elements are stronger in proportion to their size as their size decreases.

NOISE AN OBJECTIONABLE CHARACTERISTIC

Noise tends to make itself evident with increase of speed and puts an additional premium on small cylinder dimensions. Non-uniform-motion parts, especially valves and their operating mechanism, cannot be made absolutely silent, as some clearance is necessary to insure the proper seating of the valves. The production of sound tends to vary in proportion to the kinetic energy of these parts. This kinetic energy, other things being equal, varies as the square of the engine speed and inversely with the mass of the parts. The mass of the parts, if of similar form, varies as the cube of the linear dimensions. With these facts in mind, it is not difficult to understand the premium that the noise and durability problem put upon small cylinder dimensions as the engine speed is increased.

VARIATION IN FUEL CONSUMPTION A WEAKNESS

Bearing in mind our fundamental premise, that maximum desirability in the hands of the user is secured by maximum drawbar pull per pound of total weight throughout the greatest possible range of car speeds on top gear, we come face to face with the second fundamental weakness of the Otto-cycle internal combustion engine. As already pointed out the first weakness is its lack of torque at zero speed and the consequent impossibility of securing 100 per cent of torque range; the second is the fact that its fuel efficiency decreases rapidly as the load is decreased from maximum at any speed. This means, from the user's standpoint, that for any given total weight of vehicle an increase in drawbar pull (better top-gear ability in accelerating, on heavy roads and on hills) can be secured only by an increase in fuel consumption per mile of average driving.

Herein undoubtedly lies the engineer's greatest problem and greatest opportunity in motor-car design. Reduction in total road weight is the only kind of weight reduction that really counts. There is relatively so much greater opportunity to reduce total road weight in other parts of the car than in the engine, that the increase of torque range in the engine is considerably more important than the reduction of engine weight. The problem of giving the user a car of desirable driving characteristics without excessive fuel consumption, and incidentally excessive tire cost, involves an inter-relation between the design of the engine and the design of the rest of the car structure.

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In considering relative fuel consumption of cars, it is no adequate comparison to state that one car will run 12 and another 15 miles on a gallon. The car that will run the 15 miles might have on top gear a maximum acceleration rate of only, let us say, 1.6 ft. per sec. per sec., with the ability to climb a 5-per cent grade. The car that will run only 12 miles to the gallon might have an acceleration rate on top gear of 3.2 ft. per sec. per sec., with the ability to climb a 10-per cent grade. On this basis the car showing the less mileage per gallon would have the greater fuel economy in proportion to its driving characteristics.

FINAL POSSIBILITIES OF INCREASED TORQUES

It is interesting to consider what are the ultimate possibilities of torque value in proportion to total car weight and its effect upon acceleration and hill-climbing ability. Assuming weight equally distributed on steering and driving wheels and a 50-per cent coefficient of friction between the driving wheels and the road surface, we at once find the maximum acceleration rate as determined by maximum possible traction to be approximately 8 ft. per sec. per sec., corresponding to an ability to go from zero speed to 60 m.p.h. in 11 sec. Under the conditions outlined, these figures are not quite attainable because of wind resistance. On certain kinds of road surface, however, the traction effort is more than 50 per cent of the weight on the driving wheels, and the load is not necessarily equal on both driving and steering wheels. Furthermore, torque reaction within the car structure decreases the steering-wheel load and increases the driving-wheel load. Considering these modifications of the conditions originally outlined, it is well within the bounds of possibility for acceleration rates to reach the figures given. This acceleration rate would obviously correspond to a hill ability of 25 per cent (25.831 per cent as ordinarily computed), that is to a hill with an angle of something over 14 deg. 29 min. In this paper grade is considered as the ratio of vertical rise to linear distance travelled.

The relationship between acceleration and hill-climbing ability is approximate; but would be very close, if the only acceleration of an automobile were the acceleration of the entire mass in the line of travel. As a matter of fact, however, every automobile has a considerably greater ability on hills than that computed or deduced from its acceleration ability, because a considerable portion of the mass of the car has a double acceleration. All rotating and reciprocating masses are a part of the total mass of the vehicle, and are accelerated in the line of the vehicle travel. In addition they are accelerated within their several independent paths of motion, and the energy absorbed in this independent acceleration is in addition to that absorbed in the major acceleration of the whole mass.

There are, however, three great obstacles in the way of giving the average car user a vehicle with an acceleration capacity of much over 3 or 4 ft. per sec. per sec. First, the present high and increasing cost

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of the fuel; second, the need of apparently unachievable weight reduction; and third, the need of some form of prime mover that does not lose its fuel efficiency so rapidly as its load is decreased from maximum at any speed.

FORMULAS FOR CAR PERFORMANCE

I shall try to reduce the generalities of car performance to workable formulas and to suggest at least one method of determining accurately the value of a car in terms of its performance ability.

For acceleration values $32.16\ D/W = A$, in which D is drawbar pull in pounds, W is weight in pounds, and A is acceleration in feet per second per second, wind resistance being omitted.

For hill ability $D/W = G$, in which D is drawbar pull in pounds, W is weight in pounds, and G is the percentage of grade upon which the car will just maintain its speed. This percentage is expressed in terms of the linear distance traveled on the grade; that is, it is the sine of the grade angle. As an expression of average road ability, this formula should take the form of $D/W^x = A'$, in which D is the average drawbar pull in pounds throughout an agreed speed range, W is weight in pounds, including passengers and equipment, x is an exponent with a value of more than one, depending upon average road surface conditions, A' is an arbitrary value for comparing the average road ability of cars. On the assumption that two given cars have an equal value of D/W the lighter weight car will have the better road ability on soft roads. I suggest that the most accurate way of determining this value is to test a car on a level track, carrying its full equipment and passenger weight, and having, as a trailer, a towing dynamometer arranged to record the drawbar resistance at all speeds throughout the desired range. An average value of D can thus be determined for all speeds throughout an agreed range, as, for example, between 5 and 50 m.p.h. The values of A' will then express truly the comparative average road ability of a car under actual service conditions.

Having outlined in a general way the principles of a car performance and their relation to modern engine development, I shall turn to some of the specific problems that have virtually been created by the modern increase in engine speeds and in power output at these increased speeds. These two problems are lubrication and crankshaft counterbalancing.

LUBRICATION OF HIGH-SPEED ENGINES

In order to understand just what the problem of lubrication is and why new difficulties in its solution are met under the conditions outlined, we must bear in mind that in ordinary driving on level roads the duty of the engine is simply that of maintaining the desired car speed. The modern type of engine is operating at somewhat higher speeds and under much less load than those in the older types of car with lower top-gear acceleration rate and less hill ability.

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Consequently, the opportunity for over-lubrication, with its attendant difficulties (carbon deposits in the combustion chamber and between exhaust valves and their seats, foul spark-plugs and excessive oil consumption) is increased. This difficulty is further aggravated by the need at top speed and full load of greatly increased lubrication over that necessary in the older (lower-torque lower-speed) engines.

In all of the older systems with which I am familiar, including both the conventional force-feed system and the splash system, the oil lubricating the crankshaft and crankpin bearings must perform a two-fold function; it must lubricate the bearings and also, in part at least, cool these bearings.

This outline of modern conditions brings us squarely up to the problem of how an engine should be lubricated so that for the major part of the time it can be run at moderate speeds with light loads without over-oiling, and be run at any time at relatively high speeds, developing a relatively large power output without being under-lubricated and without permitting the temperature of the crankshaft and crank-pin bearings to rise beyond that at which the oil retains its lubricating properties. It is general practice to arrange engine oiling systems so that an increase in speed automatically increases the amount of lubrication. That this practice has not been entirely satisfactory is not surprising, when we consider that with a car traveling at say 25 m.p.h. the lubrication will be the same whether the engine merely has to keep the car rolling on a cement road or is working at its maximum power output at that speed, as in pulling the car through a heavy road or up a steep grade.

There is no dodging the issue. Variation of lubrication, in proportion to engine speed only, means under-oiling under some conditions or over-oiling under other conditions, or perhaps some of both, and the wider the torque range of the engine and the greater its torque capacity per cubic inch of piston displacement the greater the error in this form of lubrication control.

In a somewhat less common form of lubrication control, but I believe a better and a growing one, the oil control is interconnected with the engine throttle, so that as the throttle is opened to increase the torque output of the engine, the lubrication is increased to meet the increased need of lubrication. This interconnection of lubrication has been adapted successfully to both splash and force-feed oiling systems.

COMMON WEAKNESS IN OILING SYSTEMS

For use in high-speed high-power engines, the oiling systems in general use have one common weakness, namely: the oil for the crankshaft and crankpin bearings must, in part at least, cool as well as lubricate these bearings. In engines with cylinders and pistons of proper clearance and proper finish the margin is exceedingly narrow between over-oiling of cylinders and under-oiling of crankpin

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and crankshaft bearings. The accuracy of this statement is shown by the number of modern high-speed engines equipped with scraper rings and other devices for keeping the oil down in the cylinders. Such devices are used in spite of the fact that in most, if not in all, good lubricating systems, the cylinders receive but a small part of the oil that has passed through the corresponding crankpin bearing.

Since the cylinders do not, or at least should not, get any oil, except a small percentage of that which has already passed through the corresponding crankpin bearing, I believe that under-lubrication would always show itself first between pistons and cylinders if it were not for the fact that the oil passing through the crankpin bear-

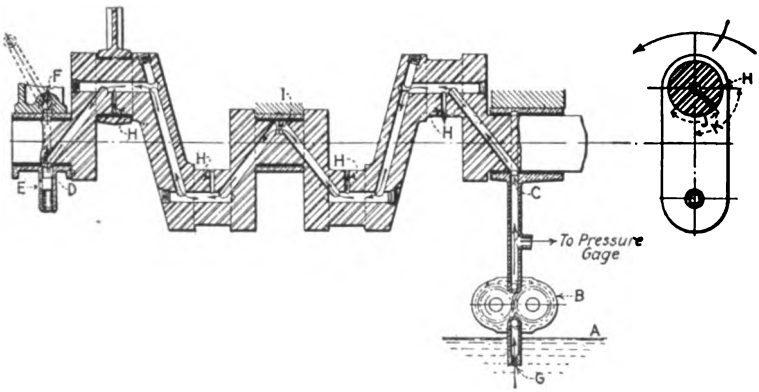

FIG. 1—MAIN ELEMENTS OF FORCE-FEED OILING SYSTEM

ings must cool as well as lubricate these bearings. If this double duty of the oil in crankshaft and crankpin bearings can be eliminated successfully, I believe that it will be entirely feasible to do away with scraper rings and similar devices (which after all obviate only in part the troubles due to cylinder over-lubrication) and work out the control entirely from the standpoint of correct cylinder lubrication for the various loads and speeds.

Fig. 1 shows a diagrammatic layout of the main elements of a force-feed oiling system, which it seems to me is one method of meeting the oiling problem, as outlined, in its entirety. Fig. 2 shows the same system applied to a two-bearing crankshaft for eight-cylinder V-type engines, with side-by-side connecting-rod bearings, similar parts being similarly lettered. Fig. 2 also shows a shaft of this type correctly counterbalanced for high speed. This drawing will be used later for consideration of the general subject of counterbalancing.

In this oiling system the pump delivers its full capacity of oil through an oil line reaching every bearing on the crankshaft. Every

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one of these bearings is positively reached by the oil, even though no pressure is registered on the oil gage. When pressure is carried on the oil line the pressure is equal on every bearing, except as modified by centrifugal effects and friction of flow within the oil line. The flow of oil throughout the length of the crankshaft will maintain automatically the temperature of all crankshaft bearings closely approximate to that of the main body of the oil in the reservoir. The oil used for lubricating the bearings has no other duty to perform. If, as should always be the case, the crank-throw cheeks adjacent to the ends of the center line bearings are finished with square corners in planes outside the cylinder diameters, no oil thrown off from these center line bearings will reach the cylinder. The oil

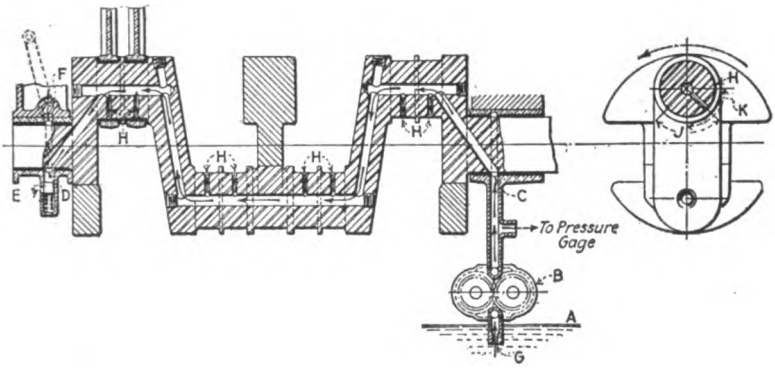

FIG. 2—FORCE-FEED OILING SYSTEM APPLIED TO TWO-BEARING CRANKSHAFT

for cylinder lubrication will, as is proper, be only a small percentage of the oil that has already passed through the crankpin bearings. In this way it can be assured that the oil will be so controlled that the cylinders will never be over-lubricated, and that at the same time all crankshaft and crankpin bearings will be lubricated adequately *and their temperatures controlled properly.*

It is common knowledge to those familiar with the problem, that little or no pressure is required on so-called pressure oiling systems to lubricate properly an engine running idle or under light loads; but with pressure systems as ordinarily constructed and controlled, there is a possibility that some or all of the bearings will not receive oil unless the oil gage registers some pressure. A glance at Figs. 1 or 2 will show that the stated form of pressure or force-feed oiling assures oil delivery to every bearing, even though the outlet *F* is wide open and no pressure is shown on the oil gage. The normal outlet-valve *F* interconnected to the engine throttle is entirely closed whenever the throttle is wide open; that is, whenever the engine is delivering its maximum torque output, the oil pressure will

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be up to the point determined by the spring control of the pressure outlet *E*.

MAXIMUM CRANKPIN BEARING PRESSURES

I should define a high-speed Otto-cycle internal-combustion engine as an engine in which the maximum crankpin bearing pressures are due to the inertia and centrifugal forces set up by the piston and connecting-rod. In a high-speed engine as described, the maximum crankpin bearing pressure occurs on the inner side of the crankpin. In such an engine the oil delivery point to the crankpin bearing should be within and probably at about the mid-point of the arc *K*, Figs. 1 and 2, for the reason the oil delivery point will have passed over that portion of the connecting-rod bearing subjected to maximum pressure, immediately before the occurrence of such maximum pressure, thus assuring the maximum amount of oil within the zone of maximum pressure in proportion to the total amount of oil delivered through the bearing.

Figs. 1 and 2 represent single-plane crankshafts as used in four-

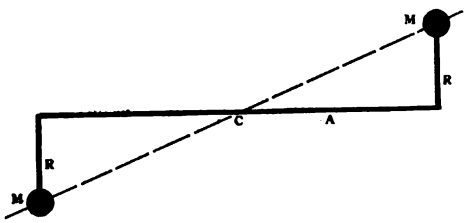

FIG. 3—STATIC BALANCE OF ROTATABLE MASSES

or eight-cylinder engines, but it is obvious that the arrangement is equally applicable to three- or four-bearing crankshafts of the three-plane type, as used in six- or twelve-cylinder engines.

CRANKSHAFT BALANCING AND COUNTERBALANCING

We now come to a phase of engine design, the effect of which increases as the square of the engine speed. In any rotating mass three kinds of balance are involved. For comparison and analysis I shall call them static balance, running balance and distortion balance.

Static balance is secured whenever the axis of a rotatable mass passes through its center of mass, but such a mass need not necessarily be in either running or distortion balance.

Any mass in either running or distortion balance must however be in static balance; that is, running balance and distortion balance are each a special condition or form of static balance.

Running balance can be described as that condition or form of

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static balance in which any centrifugal couples are equal and opposed and therefore neutralize each other.

Distortion balance can be described as that condition or form of *static and running balance* in which all centrifugal forces within any plane at right angles to the axis (that is, within any plane of rotation) are balanced.

For purposes of illustration and analysis, let us consider the diagrams of rotatable masses, Figs. 3, 4 and 5, in which A represents the axis, M any specified mass, R any specified distance between M and the axis, and C the center of mass of the system.

In Fig. 3 we have a case of static balance, inasmuch as A passes through C, but we have not running balance, since M and M' upon rotation about A will form an unneutralized centrifugal couple tending to force A out of position.

In Fig. 4 we have a condition of static balance (inasmuch as A passes through C) that is also running balance; that is, the centrifugal

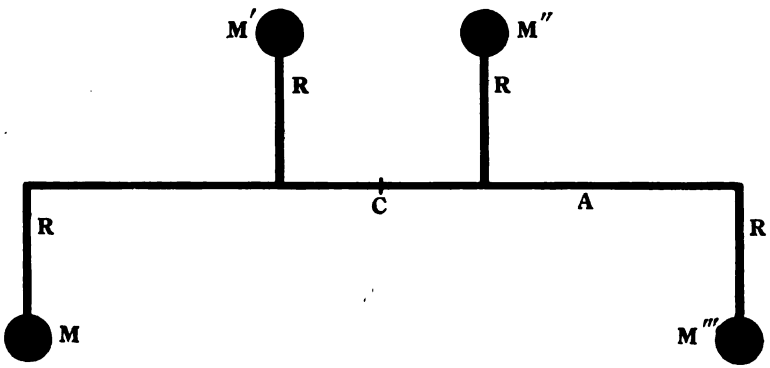

FIG. 4—STATIC AND RUNNING BALANCE OF ROTATABLE MASSES

couple between M and M' is equal and opposed to the centrifugal couple between M'' and M''', and there is no tendency to displace A as a whole. Fig. 4 does not, however, represent a condition of distortion balance, as the opposed centrifugal couples tend to bend the axis A.

If, however, our masses be arranged as in Fig. 5, so that M' and M lie in the same plane of rotation as do also M'' and M''', we have the special form or condition of static balance (inasmuch as A passes through C) that also represents running and distortion balance.

BALANCE IN FOUR- AND SIX-THROW CRANKSHAFTS

As applied to conventional four- and six-throw crankshafts, static and running balance are, or should be, always achieved within the limits of workmanship.

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For reasons I shall point out later, a single-throw crankshaft, for best results in service, should not be in either static or running balance; a two-throw crankshaft with the throws opposed, for best results in service, should be in static, but not in running balance.

It is only within the present period of high-speed engine developments that the importance of distortion balance, so nearly as it can

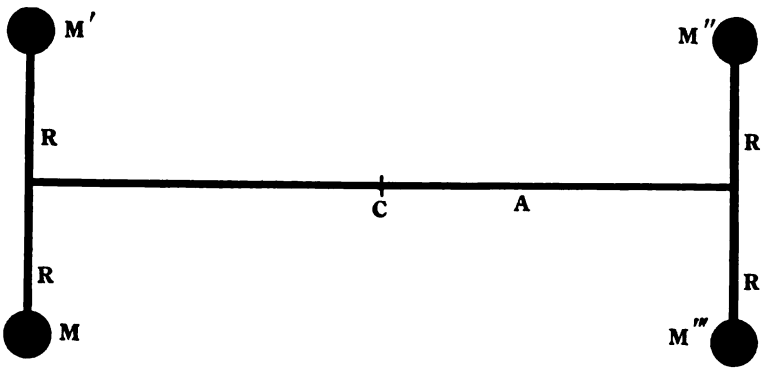

FIG. 5—STATIC BALANCE THAT ALSO REPRESENTS RUNNING AND DISTORTION BALANCE

be achieved in a crankshaft, has begun to be understood. Inasmuch as its functions exercise a certain control over the form of a crankshaft, what is commonly known as counterbalancing is required to bring any crankshaft into any condition of static balance.

Running balance in conventional four- and six-throw crankshafts, by means of counterbalancing can readily be and, as I have said, is almost universally secured. Distortion balance in a crankshaft can

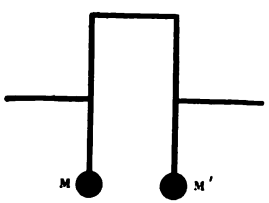

FIG. 6—COUNTERWEIGHTED SINGLE-THROW CRANKSHAFT

never be absolute, because of the effect of its functions upon the form of the shaft. The closest possible approximation to distortion balance that can be secured in any crankshaft was commonly used before the advent of the automobile.

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IRREDUCIBLE MINIMUM DISTORTION COUPLE

Fig. 6 represents a single-throw crankshaft in which M and M' are counterweights, having a common centrifugal effect equal to the centrifugal effect of the crank-cheeks and -pin. There is obviously a slight distortion tendency here, due to the fact that the pin mass lies between the counterweights M and M', with a slight resultant tendency to bend the crankpin. This approximation of distortion balance in a crankshaft can be properly described as the irreducible minimum or single-throw distortion couple, and will be recognized as ancient mechanical practice.

I shall hereafter refer to a crankshaft which has a single-throw distortion couple, as being in complete distortion balance. Of course

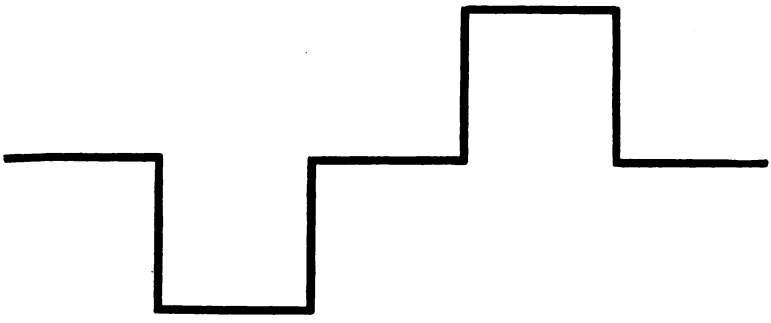

FIG. 7—SINGLE-PLANE TWO-THROW CRANKSHAFT IN STATIC BALANCE

no condition of static balance can be achieved in a single-throw crankshaft without resorting to the addition of counterbalancing masses, as represented in Fig. 6. In Fig 7 however (representing a single-plane two-throw crankshaft) we find a condition of static balance (not of running balance) apparently without the use of counterbalancing. The counterbalancing exists, inasmuch as one throw acts as a counterbalance for the other. This is also true in Fig. 8, which represents the conventional three-bearing four-throw four-cylinder crankshaft. In Fig. 8 however (which like Fig. 7 is automatically counterbalanced without the addition of extra counterbalancing masses) we have a natural condition of running balance, but not complete distortion balance, and the center bearing is required to help resist the bending tendency due to the lack of distortion balance. This load on the center bearing, while negligible in low-speed engines, is by no means negligible in the modern high-speed engines, and it is not unusual to see counterbalancing masses added to shafts of this type, as in Fig. 9, to bring them to complete distortion balance.

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TWO-BEARING FOUR-CYLINDER CRANKSHAFTS

In small-bore high-speed engines the lack of distortion balance causes much greater load upon the center bearing than does the explosion pressure in the cylinders. This fact accounts for some of the troubles encountered in two-bearing four-cylinder crankshafts. Such crankshafts are feasible for small-bore high-speed engines, provided counterbalancing masses are added to the crankshaft as indicated in Fig. 10. In this figure the value given for the center

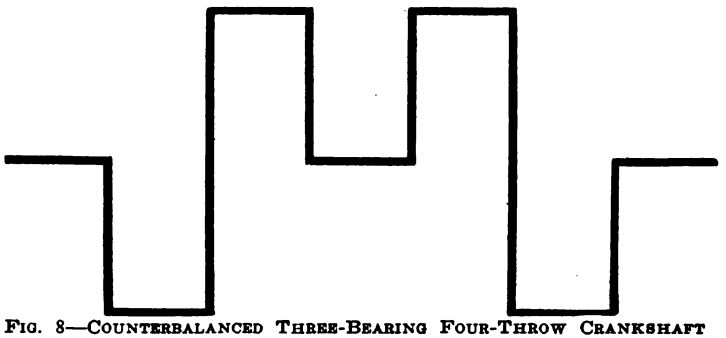

FIG. 8—COUNTERBALANCED THREE-BEARING FOUR-THROW CRANKSHAFT

counterbalancing mass is $2M+$, since the center crankpin carrying two connecting-rods is always longer than the two end pins. This difference in length and weight should be considered in determining the centrifugal value of the center counterbalancing mass.

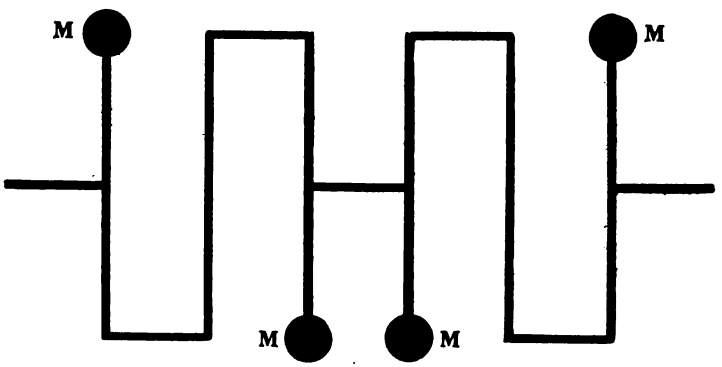

FIG. 9—CRANKSHAFT AS IN FIG. 8, WITH ADDITION OF COUNTERBALANCING MASSES

In both Figs. 9 and 10 the distortion couple is reduced to the minimum of a single throw, although a counterbalancing mass is added only at one side of each throw. A counterbalancing mass might be added at each side of each throw, as shown in Fig. 6, but inasmuch as the crankshafts so far considered are all single-plane crankshafts (that is crankshafts in which the axes of all crankpins

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and the axis of the crankshaft lie in a single plane) the mathematical merging of two counterbalancing masses, opposed and in the same plane of rotation, results in a cancellation of their centrifugal forces and therefore *permits* the elimination of such masses. So far as the single-plane crankshaft is concerned, the securing of complete triple balance (that is, static and running balance with distortion couples reduced to the irreducible minimum of a single throw) is mathematically identical with the counterbalancing of a single-throw crankshaft.

SIX- AND TWELVE-CYLINDER ENGINE CRANKSHAFTS

Let us now consider the three-plane crankshaft as used in modern six- and twelve-cylinder automobile engines. Fig. 11 shows a side and an end view of a counterbalanced single-throw crankshaft, in which,

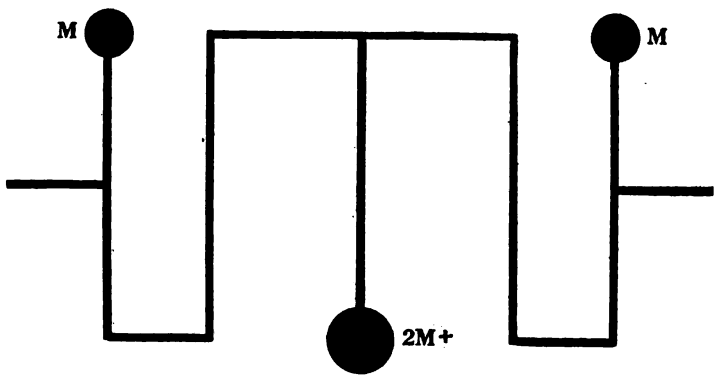

FIG. 10—TWO-BEARING FOUR-CYLINDER SHAFT WITH COUNTERBALANCING MASSES

as is obviously possible, the applied counterbalancing mass has a contour such that the included angle between its sides when viewed from the end is 120 deg., P representing the center of crankpin, A the axis of the crankshaft, and M the center of mass of the counterweight. If six of these throws are formed into one shaft with a bearing on each side of each throw, we can obviously have a conventional seven-bearing six- or twelve-cylinder crankshaft in complete triple balance.

If we omit a bearing between any two throws, one cheek and one counterbalance of each of these throws will lie in the same plane, as illustrated in the end view in Fig. 12. In this type of shaft the angle between each pair of adjacent crankpins, except the two center ones, is 120 deg. As in Fig. 11, P represents the center of the pin, A the axis of the shaft, and M and M' the centers of mass of the counterweights. Inasmuch as the specified included angle between the sides of each counterweight is 120 deg., these two counterweights

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when brought into the same plane by the elimination of an intermediate bearing, will have their adjacent edges in contact as shown along the line *C*. This means that the two counterweights can be merged into one having an included angle between the sides equal to the sum of the included angles of both weights, with the common center of mass lying in the line of union *C* of their adjacent sides, and having a mass equal to the sum of the two counterweights. Any form of counterweight can be used whose center of mass lies in this same line (*C* in Fig. 12), the centrifugal value of which is the same as the combined counterweights shown. For obvious mechanical rea-

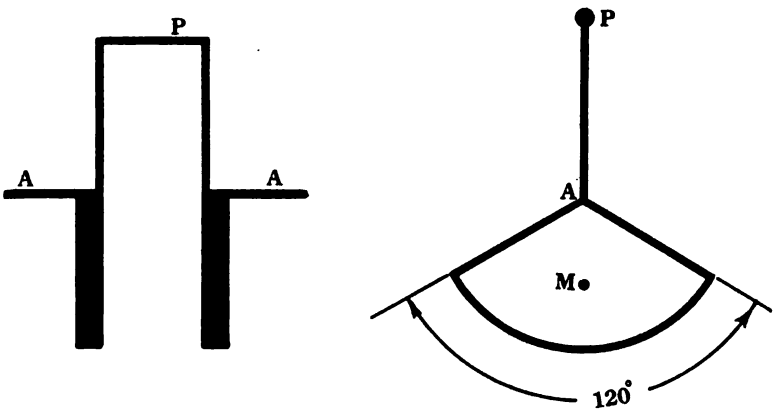

FIG. 11—SIDE AND END VIEWS OF COUNTERBALANCED SINGLE-THROW CRANKSHAFT

sons it will always be desirable to adopt a counterweight form with an included angle of less than 240 deg. between its sides, but the adoption of such a different form of counterweight in no way alters the mathematical fact that it is and must be the exact centrifugal equivalent of the two combined counterweights, as illustrated in Fig. 12.

SIX-THROW SHAFT WITH TWO CENTER THROWS

Now let us consider the special case of the two center throws in the conventional three-plane six-throw crankshaft. Fig. 13 shows these two center throws in the seven-bearing form of this type of crankshaft (that is, with a bearing on each side of each throw), in which *A* represents the axis of the crankshaft, *P* the center line of the pins, and *M*, *M'*, *M''* and *M'''* the counterbalancing masses. If, as in the four-bearing conventional three-plane six-throw crankshaft, the center bearing be omitted, this central portion of the shaft takes the form illustrated in Fig. 14, in which case complete triple balance

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can be secured for these two center throws with three counterbalancing masses instead of four, the center counterbalancing mass of the three obviously having a value of *2M*. An expedient that secures a close approximation to distortion balance is shown in Fig. 15, in which only two counterbalancing masses are required, each having the value of *2M*. In this case the distortion couple is two throws long and therefore not the minimum, but is a sufficiently close approximation for all practical purposes.

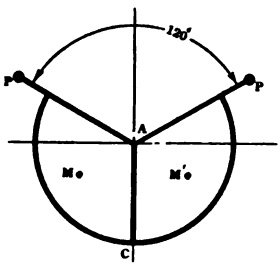

FIG. 12—END VIEW SHOWING CHECK AND COUNTERBALANCE IN PLANE

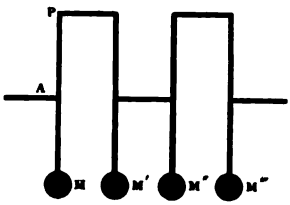

FIG. 13—PART OF THREE-PLANE SIX-THROW CRANKSHAFT

Fig 16 shows a still further simplification (which has been used) of securing approximate triple balance of the two coincident center throws of a four-bearing three-plane six-throw crankshaft, in which these two center crankpins are counterbalanced by the use of a single counterbalancing mass, having a value of *4M*. The conventional four-bearing three-plane six-throw crankshaft in Fig. 17 was designed

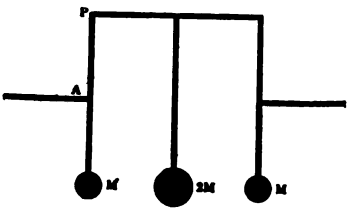

FIG. 14—PART OF THREE-PLANE SIX-THROW CRANKSHAFT WITH CENTER BEARING OMITTED

under the author's supervision early in 1912. Approximate triple balance was secured by merging the three center counterbalancing masses in Fig. 14 into one, as shown in Fig. 16, and also by merging the three counterbalancing masses, theoretically necessary for each of the two end throws, into two counterbalancing masses.

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To secure what I have described as complete triple balance in conventional three-plane six-throw crankshafts, the seven-bearing type requires twelve counterbalancing masses, and the four-bearing type obviously requires nine counterbalancing masses, in which case each of two of these masses is the centrifugal equivalent of two 120-deg. merged weights, and one of them is a coincident merged weight as in Fig. 14, the others being simply single-throw counterbalancing masses. The four-bearing crankshaft, with the exception of the center throws, can, however, be in complete triple balance with eight weights.

The conventional three-bearing three-plane six-throw crankshaft is in some ways the simplest crankshaft of this type in which to secure what I have described as complete triple balance. In this case only eight weights are required; that is, four single-throw weights on the cheeks adjacent to ends of the center line bearings and four 120-deg. merged weights on each of the four merged crank-throw arms where center line bearings are omitted.

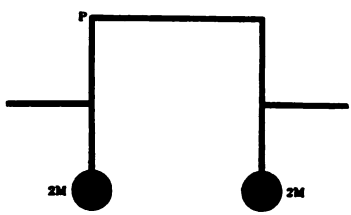

FIG. 15—METHOD OF SECURING APPROXIMATE DISTORTION BALANCE

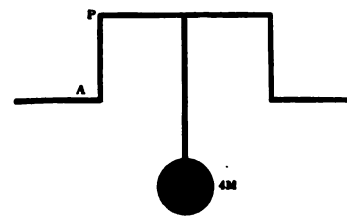

FIG. 16—APPROXIMATE TRIPLE BALANCE IN SIX-THROW SHAFT

SINGLE-THROW SHAFT BASIS OF BALANCING

The foregoing description of fundamentals shows that the science of counterbalancing a crankshaft of any number of throws, or of any form, is only the science of counterbalancing a single-throw crankshaft, inasmuch as the mathematical merging of two counterbalancing masses lying in the same plane of rotation is an obvious mechanical expedient. Bearing this in mind, it is evident that we can examine the whole subject by analyzing correct counterbalanced values for a single-throw crank. In the remainder of this discussion, I shall therefore confine myself to the single-throw form of crankshaft.

We have so far considered the counterbalancing problem as regards the crankshaft itself. This is not sufficient, however, if counterbalancing is to achieve the desired result of minimizing pressure on the center line crankshaft bearings due to tendencies to distort under service conditions, especially in high-speed engines. In the first place, the centrifugal effect of the crankpin and crank-cheek mass is increased by that portion of the lower end of the connecting-rod immediately surrounding the crankpin. In addition the centrifugal effect

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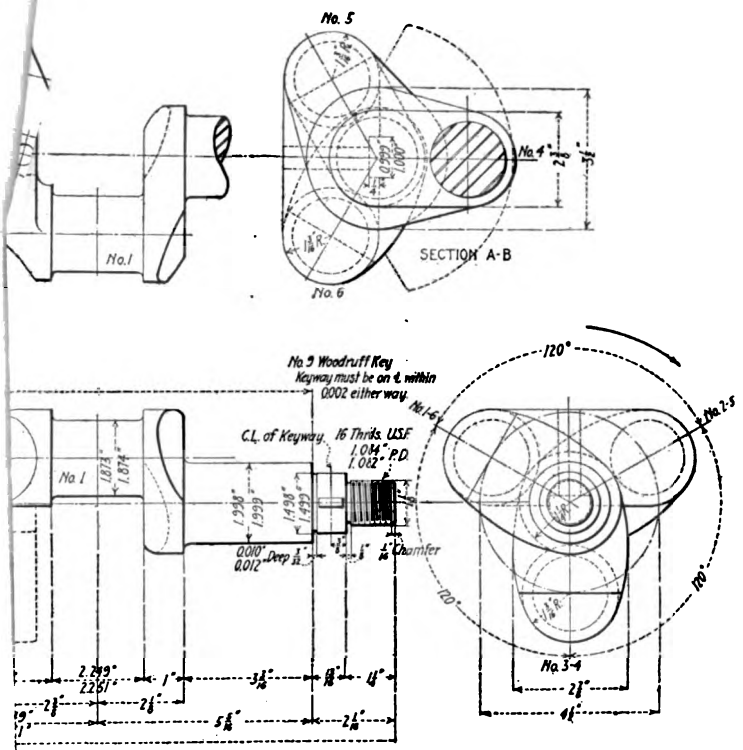

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of the intermediate portion of the connecting-rod supplements that of the crank-cheeks, the crankpin and lower end of the rod in distorting the crankshaft; and further, the centrifugal effects of the upper end of the connecting-rod and the reciprocating mass of the piston and piston-pin during certain portions of each rotation of the crankshaft also tend to distort the crankshaft. All of these tendencies combined can be lessened by the selection of proper counterbalancing masses.

In order to secure the minimum tendency to distort the crankshaft, the counterbalancing mass must be greater than that necessary properly to counterbalance or neutralize the centrifugal effects of the crank-cheeks and crankpins. This is merely another way of saying that a crankshaft that alone is in the nearest triple balance possible will not have the minimum tendency to distort in service; and that a crankshaft properly counterbalanced, so as to have the minimum tendency to distort in service, will show a marked tendency to distort when run at high speed without the connecting-rods and pistons attached.

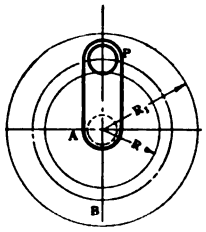

FIG. 18—END VIEW OF SINGLE-THROW CRANKSHAFT

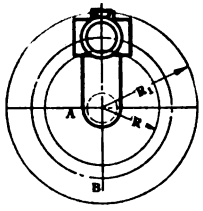

FIG. 19—SHOWING LOWER END OF CONNECTING-ROD

CORRECT COUNTERBALANCING MASSES

I shall close my discussion of crankshaft counterbalancing with a brief consideration of what are correct counterbalancing masses to permit the least tendency to displace the axis of any crankshaft under service conditions. As I have already indicated, when a method of determining these values for a single-throw crankshaft is worked out, the correct application to a crankshaft of any number of throws involves nothing but simple and obvious mechanical and mathematical merging of those crank-cheeks and counterbalances that, owing to the omission of center-line bearings, lie in the same planes of rotation. Fig. 18 represents an end view of a single-throw crankshaft, in which A is the axis, P is the center of the crankpin, and R is the uniform radius, representing the uniform radial pull tending to displace the axis at any given rate of speed, and always acting in a plane passing through the axis A and the center of the pin P. If suitable masses whose centers of mass lie within the plane B are arranged as indicated

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in Fig. 6, R will be opposed by an exactly equal force acting in the opposite direction, and its value will become zero. Considering Fig. 19, which is identical with Fig. 18, except that the lower end of the connecting-rod immediately surrounding the crankpin has been indicated, we see that the value of R has been increased to some new value as R_1, and that the centrifugal force due to our counterbalancing masses must be increased correspondingly to eliminate all tendency to displace the axis.

In Fig. 20, representing the mechanical train, that is, the piston and connecting-rod, attached to the crankpin, we find that the value

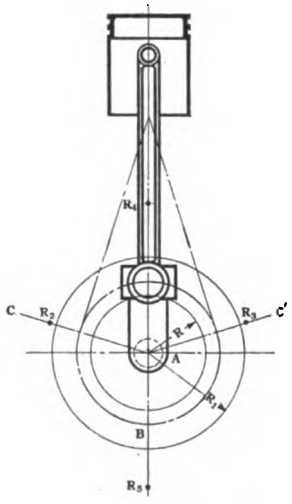

FIG. 20—PISTON AND ROD ATTACHED TO CRANKPIN

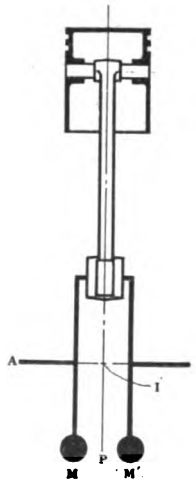

FIG. 21—COUNTERBALANCED MECHANICAL TRAIN

of R_1 is increased and thrown into variable form, and that this variable form has two points of minimum value, as at R_2 and R_3, lying in the radial lines C and C' with radial values slightly greater than R', the lines C and C' making right angles with the center line of the connecting-rod when the center of the crankpin lies within C and C'. As the crank in its rotation approaches either of the two dead centers from the points R_2 and R_3, the resultant forces tending to displace A are increased, having (owing to the finite length of the connecting-rod) a maximum value at the upper dead-center, as represented at R_4, and having still another value less than R_4, but greater than R_2 and R_3 at the lower dead-center, as indicated at R_5.

It now becomes evident that if through the addition of counterbalancing masses we are to secure the minimum tendency to dis-

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place the axis A, the centrifugal value of the counterbalancing masses must be greater than R_2 and R_3 and less than R_5. Since revolving counterbalancing masses attached to the crankshaft can have only a constant radial value for any given speed, it is at once seen that in actual operation counterbalancing cannot entirely eliminate all tendencies due to the motion of the parts to displace the axis A. There is, however, a counterbalancing value for every crank throw and for its mechanical train, which under service conditions will permit a minimum tendency to displace A.

In Fig. 21, A is the axis of the crankshaft, P is a plane of rotation about A, passing through the center line of the mechanical train and I is the intersection of A and P. At the point I the motion of parts sets up tendencies to displace A, and these tendencies can, as we have already seen, be reduced greatly by the selection of proper values of centrifugal force for counterbalancing masses indicated at M and M', whose resultant centrifugal effect, owing to their symmetrical arrangement on each side of the plane P, lies within the plane P.

No matter how many throws or what the form of a crankshaft, there will be a point I for every throw, and the resultant tendency to displace A are increased, having (owing to the finite length of the the number of throws or their angular relation. It is entirely immaterial, so far as these effects are concerned, whether center-line bearings are interposed between the throws.

AUTHOR'S CONCLUSION

The analysis given is sufficiently complete for the practical purpose of computing the proper centrifugal values for counterbalancing masses for all conventional types of engines in which only one piston and connecting-rod are attached to a single crankpin.

With V engines it is also true that the entire problem of correct counterbalancing is involved in the determining of correct counterbalancing values for a single crank-throw.

I have not attempted to exhaust the subjects under discussion, and have omitted entirely several interesting phases of high-speed engine design, as, for example, the analysis of the valving train, various installations of electrical and other auxiliary equipment, carbureter location, and intake manifold sizes and forms as affected by increased engine speeds.

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DYNAMICS OF VEHICLE SUSPENSIONS

By Dr. Benjamin Liebowitz

(Member of the Society)

Abstract

Believing that a knowledge of definite, quantitative relations affecting the behavior of vehicle suspensions will lead to a better understanding of the problems of spring design, the author analyses the problem by considering it as a simple system (a single wheel supporting a weight by means of a frictionless spring), and by assuming that this system encounters certain elementary forms of irregularities.

Analysis of this problem results in several expressions, the interpreting of which reveals the influence which the varying of specific factors has upon the action of the spring and its suspended load. These expressions take into account the effect of the ratio between sprung and unsprung weight, of speed, of size and kind of irregularity encountered by the wheel, tire inflation, flexibility of the spring, and wheel diameter. Other factors which are considered as a result of interpreting the curves presented are the effect of friction in the suspension, and of synchronism. Statements of the betterment which can be looked for in riding and steering qualities as a result of varying different factors are made. The effect of shock absorbers is discussed.

I—INTRODUCTION

That the riding qualities of our present motor vehicles are unsatisfactory is believed by many people both in and out of the automobile industry. As to what the causes of this unsatisfactory condition may be, however, there is great diversity of opinion, as illustrated by the variety of improvements which have been suggested. The difficulty is that few engineers have a sufficiently clear understanding of what happens when a wheel encounters an irregularity, what path the wheel takes in getting over the obstruction, what influence weight distribution and wheel diameter and character of the suspension may have and what forces would be impressed on the body *if the suspension were ideal.* When these matters are clear we shall know what the behavior of springs *ought* to be, and by comparing this with their actual behavior we shall understand the shortcomings of our present suspensions. Then, as always, when the cause is known, the cure will be simplified.

The object of this paper is, first, to investigate theoretically the actions which take place when an irregularity is encountered and to determine the influence of the various factors, such as weight dis-

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tribution, etc. This will be followed by a brief discussion of the evidence bearing on the supposed shortcomings of present suspensions.

II—STATEMENT OF THE PROBLEM

In order to make the subject amenable to analysis and the results simple enough to be useful, it is necessary to introduce approximations and limitations. These will not destroy the usefulness of the theory, however, for even though not mathematically exact, the results will apply to most cases with sufficient accuracy for all practical purposes.

The first step in the simplification of the problem is to restrict ourselves to one wheel only, whose weight may be taken as roughly equal to one-fourth the unsprung weight of an actual motor vehicle. Mounted on this wheel by means of a spring suspension is a weight or

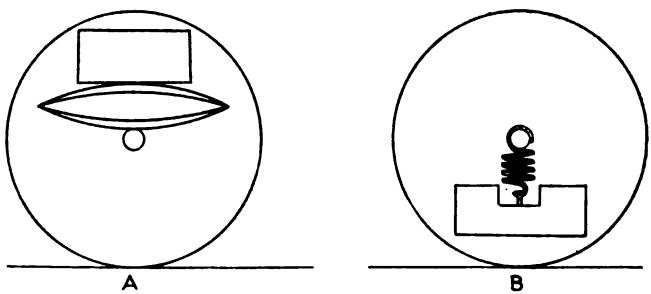

FIG. 1—TYPES OF MATHEMATICAL SPRING SUSPENSIONS

"body" whose center of gravity we shall suppose to be directly under or directly over the center of the wheel. This weight may be taken as roughly equal to one-fourth the sprung weight of the actual motor vehicle, with or without "live" load, as the case may be. The mathematical part of our theory will deal, therefore, with a system such as shown in Fig. 1, *A*, or *B*.

At first we shall regard the spring suspension as "ideal," that is, devoid of friction, hysteresis, after-effect and inertia. Departures from these ideal conditions will then be taken up, particularly with reference to the shortcomings of our present suspensions. For the sake of simplicity we shall always assume that the load deflection curve of the suspension is a straight line.

We must next limit ourselves to certain forms of irregularities. There are, of course, an infinite variety of bumps, but they may be divided roughly into two classes, "sharp," and "flat" or "rolling." An irregularity which rises suddenly out of a level roadbed, for example, a stone, may be called a "sharp" bump; for such the deformability of the tread plays a dominant role. An irregularity which has

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a more or less gradual approach, like those shown in Figs. 2 and 3, may be called "flat" or "rolling"; for such the deformability of the tread is of little importance. Most bumps are partly "sharp" and partly "flat" according to these definitions, but it is sufficient to treat only the extreme cases, for when these are clearly understood the behavior toward average bumps will be likewise understood.

As a "sharp" bump is pressed further and further into the tire, the resisting force becomes greater and greater. If the bump has a simple shape, the relation between the indentation and the force producing it will be approximately a straight line. We shall always assume this to be the case in our treatment of sharp bumps.

In dealing with this type of irregularity a great saving of labor results if we restrict ourselves to bumps which are not greater in height than one-ninth or one-tenth of the wheel radius. For wheels of ordinary diameter, therefore, we shall not consider obstructions more than two inches or so in height. In the case of "flat" bumps, however, this restriction is not made.

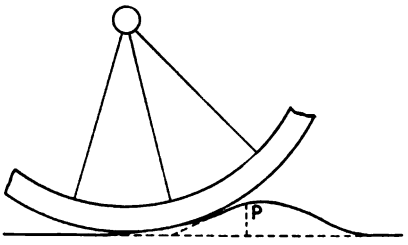

FIG. 2—WHEEL MEETING "FLAT" BUMP

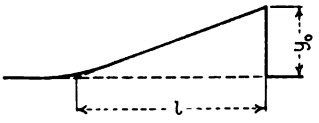

FIG. 3—SIMPLIFIED "FLAT" BUMP

Certain assumptions about the speed of the vehicle must be made. No upper limit to the speed need be imposed, for the theory will hold for velocities as high as we can get, but some lower limit must be set. At very low speeds, the vehicle as a whole simply follows the contour of the roadbed, more or less smoothed by the deformability of the tread; the suspension plays an insignificant part. The phenomena in which we are interested do not appear generally until speeds of from 8 to 12 m.p.h. are attained. The theory as developed herein will hold with a fair degree of accuracy for most vehicles for speeds no lower than about 12 m.p.h.

As a consequence of the wheel meeting the obstruction, the vehicle is retarded. This is because the force due to the obstruction acting on the wheel is radial, and therefore has a fore-and-aft component as well as a vertical component. The vertical force throws the wheel upward, and its influence on the body is determined by the resulting flexure of the spring; the fore-and-aft force acts directly on the body through the spring-brackets or radius-rods. It is of primary

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importance to determine the magnitude of these horizontal forces, and likewise the amount of the retardation. If the bump is not too large, however, the horizontal component of the blow on the wheel is small compared with the vertical component, and the loss of velocity is small compared with the actual velocity. As a first approximation we can therefore neglect these small quantities, and calculate them after the wheel path has been determined.

After the wheel leaves the obstruction, it takes a flight something like that of a projectile, as will appear later. It is during this period that the displacement of the body occurs; the impact interval proper is so short that the body does not have a chance to move therein. Hence, we shall assume that the vertical motion of the body does not begin until the wheel has cleared the bump, that is, the motion of the body will be regarded as "ballistic."

The meaning of these approximations and simplifications will become clearer as the analysis proceeds. Particular stress is laid on them because in all mathematical theories of this kind, such approximations *must* be made, and it is of fundamental importance to understand clearly the limits within which the theory is applicable.

III—DERIVATION OF THE EQUATIONS OF THE TRAJECTORIES

A—*"Flat" or "Rolling" Irregularities*

In Fig. 2 a wheel is shown just as it meets a "flat" bump. In order to climb this miniature "hill" in the limited time allowed it, the wheel must attain an upward velocity. As a result of the momentum thus gained, the wheel will clear the hill at some point P just before reaching the summit. The exact location of P would have to be determined for each individual case; so, to avoid this complication, we shall take our bump to be of the simple shape shown in Fig. 3. That is, the bump is taken to be a right triangle of height y_0 and base l, except that the "approach" is rounded off, as shown. This makes the upward acceleration of the wheel during contact with the bump more or less gradual, so that the deformation of the tread can be left out of account without much error.

Just as it leaves the obstruction, then, the wheel has been displaced upward a distance y_0 and has attained an upward velocity, say v_0, which is evidently equal to $\frac{y_0}{l} V$, V being the horizontal velocity of the vehicle. The body, in the meanwhile, has also attained an upward velocity, say μ_0, which can readily be shown to be

$$\mu_0 = \tfrac{1}{2} \frac{k\, l\, y_0}{M\, V}, \tag{1}$$

where k is the stiffness of the spring and M is the mass of the body. Besides this vertical velocity, the body will also have attained a vertical displacement before the contact is over, but unless the velocity of the vehicle is low, this displacement is negligibly small.

As a result of meeting the irregularity, then, the wheel and body

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have received upward displacements and velocities, respectively, of y_o, 0, v_o, μ_o. Our problem now is to determine the subsequent motion of wheel and body until the wheel touches the road-bed again.

In addition to the symbols mentioned the following notation will be used

t = time in seconds, measured from instant wheels leaves obstruction
m = mass of wheel (weight in lb. ÷ 32)
M = mass of body (weight in lb. ÷ 32)
k = stiffness of spring (lb. per foot = 12 × lb. per inch)
x = displacement (feet) of body from its equilibrium position at any instant
y = displacement (feet) of wheel from road-bed at any instant
g = acceleration of gravity = 32
W = total weight of vehicle = $(M+m)g$
V = horizontal velocity of vehicle

When the wheel has cleared the obstruction, the only forces acting are gravity and the reaction of the spring. If the spring is ideal, as we assume, the force it exerts at any instant is equal to its stiffness times the deflection at that instant. When the vehicle is at rest, the spring is deflected X feet, let us say (all deflections will be measured in feet for purposes of analysis), and this deflection causes it to exert an upward pressure equal to the weight of the body. In other words, when everything is in equilibrium, the spring reaction equals the weight it supports. When equilibrium is disturbed due to the upward displacement of the wheel, the spring exerts an additional force equal to k times the additional deflection. This additional deflection is obviously equal to the difference between the wheel displacement and that of the body, that is, to $y - x$. Hence, the total deflection of the spring at any instant is $X + y - x$, and the total force due to the spring at any instant is

$$f = k\ (X + y - x). \tag{2}$$

This force acts *upward* on the body and *downward* on the wheel.

The only other force acting on the body is Mg, its weight, and on the wheel is mg, *its* weight. Hence, calling upward forces positive and downward forces negative, the sum of the forces acting on the body is

$$k\ (X + y - x) - Mg,$$

and on the wheel is

$$-k\ (X + y - x) - mg.$$

Equating these to the product of mass and acceleration in each case, and remembering that $kX = Mg$, we get, upon transposing:

$$\begin{cases} M\dfrac{d^2x}{dt^2} - k\ (y - x) = 0 \\ m\dfrac{d^2y}{dt^2} + k\ (y - x) = -W. \end{cases} \tag{3}$$

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The method of solving these equations can be found in any textbook on ordinary differential equations, and need not be gone into here. The solution of (3) involves four arbitrary constants, which are to be determined from the initial conditions, as follows

$$t = 0, x = 0, y = y_o, \frac{dx}{dt} = \mu_o, \frac{dy}{dt} = v_o;$$

If these constants are thus determined, we get finally

$$(4) \quad \begin{cases} x = \left(\frac{my_o}{M+m} + \frac{mMg}{(m+M)\,k}\right)(1 - \cos pt) \\ \quad - \frac{m}{(M+m)\,p}(v_o - \mu_o)\sin pt \\ \quad + \left(\frac{m}{M+m}v_o + \frac{M}{M+m}\mu_o\right)t - \frac{1}{2}gt^2 \end{cases}$$

$$(5) \quad \begin{cases} y = \left(\frac{My_o}{m+M} + \frac{M^2g}{(m+M)\,k}\right)\cos pt \\ \quad + \frac{M}{(m+M)\,p}(v_o - \mu_o)\sin pt + \frac{my_o}{M+m} - \frac{M^2g}{(M+m)\,k} \\ \quad + \left(\frac{m}{m+M}v_o + \frac{M}{m+M}\mu_o\right)t - \frac{1}{2}gt^2. \end{cases}$$

Where

$$(6) \qquad p = \sqrt{k\left(\frac{1}{M} + \frac{1}{m}\right)}$$

These equations describe the vertical motion of body and wheel from the instant the wheel leaves the obstruction to the instant it hits the road-bed again, while the vehicle as a whole is moving forward with uniform velocity V. The meaning of these equations and the important facts to be derived from them will be discussed, as will their graphs, in the next section.

A word of interpretation may be said about the quantity p appearing in (4) and (5), and defined by equation (6). The body and wheel connected by the spring constitute a system capable of vibrations. The period of vibration depends upon whether the wheel is in contact with the road; when there is no contact, the period is

$$\frac{2\pi}{p} = 2\pi\sqrt{\frac{1}{k\left(\frac{1}{m} + \frac{1}{M}\right)}}$$

That is, the quantity p, when divided into 2π, gives the period of vibration when the wheel is off the road. When, however, the wheel is in contact with the road, the period of vibration of the body is $2\pi\sqrt{\frac{M}{k}}$, provided we neglect the deformability of the tire.

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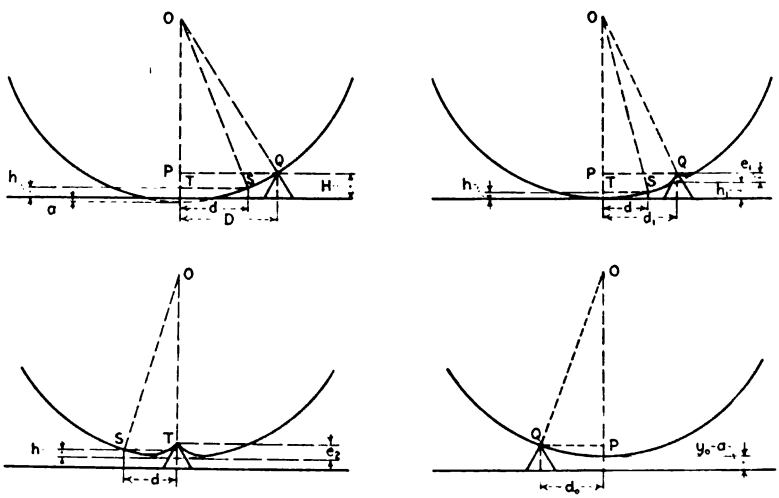
B—"Sharp" Irregularities

To investigate what happens when a "sharp" irregularity is encountered, the impact period will be divided in three stages. Stage I begins when the tread first touches the bump, as shown in Fig. 4. The tread, owing to the weight of wheel and body, is deflected or flattened at the area of contact an amount a, as indicated in Fig. 4, and now begins to rise with the wheel; when the wheel has risen a distance equal to this deflection, or distance a, the bottom of the tread will assume its normal circular shape and will just touch the road-bed, as indicated in Fig. 5. This is the end of Stage I and beginning of Stage II. Stage II continues until the center of the wheel comes directly over the apex of the irregularity, as illustrated in Fig. 6. Stage III then commences, and continues until the wheel



FIGS. 4 (UPPER LEFT), 5 (UPPER RIGHT), 6 (LOWER LEFT), AND 7 (LOWER RIGHT)—STAGES OF IMPACT OF WHEEL STRIKING BUMP

is about to leave the bump. This, as illustrated in Fig. 7, ends Stage III and the impact period.

The net result of the impact on the wheel is to raise the wheel a distance say y_0, and to give it an upward velocity say v_0; and as a consequence, the wheel takes a projectile-like flight just as in the case of a "rolling" irregularity. The equations derived in the previous part will therefore describe the motions in this case also, after the wheel has cleared the bump. Our present problem is to determine what happens during the impact period and to find the upward

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velocity v_0 and displacement y_0 of the wheel at the end of the impact period.

Referring to Fig. 4, let Q be the point of first contact with the bump, O the wheel center, whose radius is $OQ = R$, D the horizontal distance from Q to the vertical line OP, S any intermediate point on the tread between Q and the flattened portion, d the horizontal distance from S to the vertical line OP, h the vertical distance from S to the road-bed, H the height of the bump, and a the normal deflection of the tire. All these dimensions are given in Fig. 4.

From the right triangle OPQ

$$R^2 = D^2 + [R - (H + a)]^2$$

and therefore

$$D^2 = 2R(H + a) - (H + a)^2$$

But, since we shall consider small bumps (not larger than one-ninth or one-tenth of R) only, the last term may be neglected and we have

$$D^2 = 2R(H + a). \tag{7}$$

In the same manner from the right triangle OTS we find

$$d^2 = 2R(h + a) \tag{8}$$

The flattened portion of the tire pressing against the road-bed constitutes a spring, whose stiffness is $\frac{W}{a}$, for W is the total load supported and a is the deflection of the tire due to it. Experiment shows that the relation between deflection of tire and load is nearly a straight line; hence we can regard the tire against road-bed as a spring of constant stiffness k_1, where $k_1 = \frac{W}{a}$

When the wheel rises an amount y, the deflection of the tire is decreased to $(a - y)$; hence, the upward force due to the pressure of the tire against the road is, at any instant, $k_1(a - y)$.

The obstruction pressing into the tire constitutes another spring, whose stiffness depends on the nature of the tread (whether solid or pneumatic, the air pressure if pneumatic, etc.) and on the shape of the bump. If the bump has a simple shape, as shown, the force producing a given indentation will be approximately proportional to the indentation, as previously explained; hence we can regard the obstruction pressing into the tire as another spring of constant stiffness, say k_2.

The force due to the bump at any instant is, then, $k_2 \times$ indentation. To find the indentation, imagine the wheel to have moved forward until the bump is under S. (See Fig. 4.) The indentation would be $H - h$ if the wheel did not rise, but since the wheel does rise, its displacement being y, the indentation is $H - h - y$. Now, from Fig. 4, $d = D - Vt$ and, using this in connection with equations (7) and (8), we get for the indentation $\frac{1}{2R} = [2VDt - V^2t^2] - y$, and for the force due to it $\frac{k_1}{2R}[2VDt - V^2t^2] - k_2 y$.

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As already mentioned, this force has a fore-and-aft component, which, in a first approximation, may be neglected. We may assume, therefore, that the two springs k_1 and k_2 act vertically upward on the wheel.

Besides these two, there is the vehicle spring itself, of stiffness k, which acts vertically upward on the body and downward on the wheel. This force is evidently $k(X+y)$, X being the deflection of the spring due to the weight of the body, when equilibrium obtains, and y the additional deflection due to the upward displacement of the wheel. It should be noted that, just as in the case of "flat" bumps, the body is regarded as vertically at rest during the impact.

The only other force acting on the wheel is its weight mg.

Calling those forces acting upward positive, those downward negative, equating their algebraic sum to the mass times acceleration of the wheel and simplifying, we get

$$m\frac{d^2y}{dt^2}+(k_1+k_2+k)\,y=\frac{k_2}{2R}(2VDt-V^2t^2) \tag{9}$$

Since the wheel is vertically at rest when the impact starts, the initial conditions for (9) are

$$\text{when } t=0,\ y=0,\ \frac{dy}{dt}=0 \tag{10}$$

The solution of (9) subject to (10), as can be verified by substitution in (9) and (10), is

$$\begin{cases} y=\dfrac{k_2mV^2}{k'^2R}(1-\cos p_1t)+\dfrac{k_2VD}{k'R}\left(t-\dfrac{1}{p_1}\sin p_1t\right)-\dfrac{k_2V^2t^2}{k'R}, \\ \dfrac{dy}{dt}=\dfrac{k_2VD}{k'R}(1-\cos p_1t)+\dfrac{k_2V^2}{p_1k'R}\sin p_1t-\dfrac{k_2V^2t}{k'R}. \end{cases} \tag{11}$$

Where

$$\begin{cases} k'=k_1+k_2+k \\ p_1=\sqrt{\dfrac{k'}{m}} \end{cases} \tag{12}$$

If the first three terms of the well known series for sine and cosine are substituted in the first of equations (11), the following approximation is obtained

$$y=\frac{k_2V}{6m}\sqrt{\frac{2(H+a)}{R}}\,t^3\left[1-\frac{1}{4}\frac{V}{D}t-\frac{1}{20}\frac{k'}{m}t^2\right] \tag{13}$$

This approximation, which is more convenient for purposes of calculation, is valid if the speed is not too low and the tread not too stiff.

As a very rough approximation, we may put

$$y=\frac{k_2V}{6m}\sqrt{\frac{2(H+a)}{R}}\,t^3 \tag{14}$$

This shows that, roughly speaking, the motion of the wheel during the initial stage of the impact is one of uniformly increasing acceleration; that the displacement of the wheel varies directly as the velocity of the vehicle, as the stiffness of the tire against obstruction, as the square root of the height of bump plus flattening of tire, and inversely as the mass of wheel and square root of its radius. A more

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detailed study of all our equations will be given when we plot their curves.

Stage I ends, as already explained, when the wheel has risen an amount equal to the original deflection of the tire, or, when $y = a$. This, it should be pointed out, is only an approximation, because, owing to the deformation of the tire by the bump, contact between the wheel and the flat part of the road-bed will be lost before $y = a$.

The error involved is negligibly small, however, because the dominant force acting, when the impact is well under way, is the bump's pressing into the tire; so that the existence or non-existence of another point of contact makes little difference. We can say with sufficient accuracy, therefore, that Stage I ends at a time t_1 given by

$$(15) \qquad a = \frac{k_2 V}{6m}\sqrt{\frac{2(H+a)}{R}}\, t_1^2 \left[1 - \frac{1}{4}\frac{V}{D}t_1 - \frac{1}{20}\frac{k'}{m}t_1^2\right]$$

This equation can be solved by successive approximation; and by substituting the result in the second of equations (11), the corresponding vertical velocity of the wheel, v_1, can be found.

Hence, Stage I ends and Stage II begins with

$$(16) \qquad y = a, \frac{dy}{dt} = v_1.$$

The beginning of Stage II is illustrated in Fig. 5. The apex of the bump is at Q, d_1 is the distance to be travelled during this stage, S is an intermediate point on the tread, d its distance from OP, h its distance from the road-bed, and e_1 is the deformation of the tire at the instant Stage II begins. Since there is now only one point of contact, the spring k_1 is now absent. Also, since the displacement y is measured from the original position of Fig. 4, the expression for the indentation now becomes $H - h + a - y$. Also, from Fig. 5 it is clear that $d = d_1 - Vt$, $e_1 = H - h_1$, and $h = \frac{d^2}{2R}$. With these modifications we get, in a manner entirely similar to that used in Stage I, the following differential equation for Stage II

$$(17) \qquad m\frac{d^2y}{dt^2} + (k_2 + k)\, y = k_2\,(e_1 + a) - W + \frac{k_2}{2R}\,(2\,Vd_1t - V^2t^2)$$

The solution of (17) subject to (16) is

$$(18) \qquad y = a\cos p_2 t + \left(\frac{f_1}{k''} + \frac{k_2 m V^2}{k''^2 R}\right)(1 - \cos p_2 t) + \frac{v_1}{p_2}\sin p_2 t + \frac{k_2 V d_1}{k''R}\left(t - \frac{1}{p_2}\sin p_2 t\right) - \frac{k_2 V^2 t^2}{2k''R}.$$

$$(19) \qquad \frac{dy}{dt} = p_2\left(\frac{f_1}{k''} + \frac{k_2 m V^2}{k''^2 R} - a\right)\sin p_2 t + \left(v_1 - \frac{k_2 V d_1}{k''R}\right)\cos p_2 t + \frac{k_2 V d_1}{k''R} - \frac{k_2 V^2 t}{k''R}$$

In these equations

$$(20) \qquad \begin{cases} k'' = k_2 + k \\ f_1 = k_2\,(e_1 + a) - W \\ p_2 = \sqrt{\dfrac{k''}{m}} \end{cases}$$

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Stage II ends when the wheel center is directly over the apex of the bump. The total time occupied by Stages I and II is evidently $\frac{D}{V}$; hence, if we denote the duration of Stage II by t_2, we have

$$t_2 = \frac{D}{V} - t_1 \tag{21}$$

Substituting this in equations (18) and (19) gives the displacement y_2 and the velocity v_2 of the wheel at the end of Stage II.

Hence, Stage II ends and Stage III begins with

$$y = y_2, \frac{dy}{dt} = v_2 \tag{22}$$

Referring to Fig. 6, which illustrates the position of the wheel at this instant, let S be a point on the undeformed tire beyond the bump, whose apex is at T, let d be the distance from S to OT and h the distance from S to lowest tangent to the tread before it is deformed. The expression for the indentation is the same as in Stage II; that is, $H - h + a - y$, and also $h = \frac{d^2}{2R}$ approximately. But we now have $d = Vt$, so that we get for our differential equation for this stage

$$m \frac{d^2y}{dt^2} + (k_2 + k)\, y = k_2\,(H + a) - W - \frac{k_2 V^2 t^2}{2R} \tag{23}$$

The solution of (23) subject to (22) is

$$y = y_2 \cos p_2 t + \left(\frac{f_2}{k''} + \frac{k_2 m V^2}{k''^2 R}\right)(1 - \cos p_2 t) + \frac{v_2}{p_2} \sin p_2 t - \frac{k_2 V^2 t^2}{2k''R}, \tag{24}$$

$$\frac{dy}{dt} = v_2 \cos p_2 t + p_2 \left(\frac{f_2}{k''} + \frac{k_2 m V^2}{k''^2 R} - y_2\right) \sin p_2 t - \frac{k_2 V^2 t}{k'' R}, \tag{25}$$

where

$$\begin{cases} f_2 = k_2\,(H + a) - W \\ k'' = k_2 + k \\ p_2 = \sqrt{\dfrac{k''}{m}} \end{cases} \tag{26}$$

The following approximations for (24) and (25) are valid for most practical purposes

$$y = y_2 + v_2 t + \frac{1}{2m}\,(f_2 - k'' y_2)\, t^2 - \frac{1}{6}\,\frac{k''}{m}\, v_2 t^3 + \frac{1}{24}\left(y_2 - \frac{f_2}{k''} - \frac{k_2 m V^2}{k''^2 R}\right)\frac{k''^2}{m^2}\, t^4 \tag{26}$$

$$\frac{dy}{dt} = v_2 + \frac{1}{m}\,(f_2 - k'' y_2)\, t - \frac{k''}{2m} v_2 t^2 - \frac{1}{6m^2}\left(f_2 k'' + \frac{k_2 m V^2}{R} - k''^2 y_2\right) t^3 + \frac{k''^2 v_2}{24 m^2}\, t^4 \tag{27}$$

Stage III closes when the indentation is 0, that is, when $H - h - y + a = 0$. And since $h = \frac{d^2}{2R}$ and $d = Vt$, the criterion for the end of

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Stage III becomes

$$y = H + a - \frac{V^2 t^2}{2R} \tag{28}$$

Substituting this in (26) gives an equation in t, which can be solved by successive approximation. In this way we obtain the duration of Stage III, which we may call t_3. Substituting the value of t_3 in equations (28) and (27) gives us, finally, the displacement, y_0, and the upward velocity v_0 of the wheel just when the impact is over.

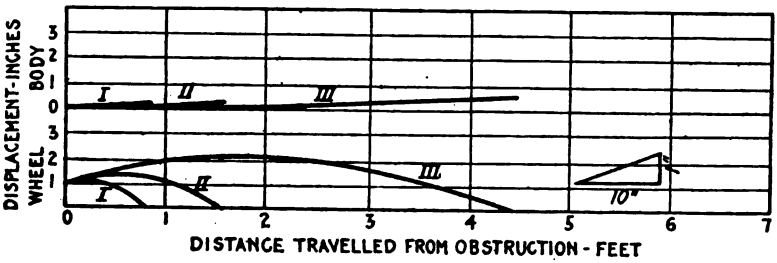

FIG. 8—WHEEL AND BODY TRAJECTORIES—EFFECT OF SPEED (Unsprung weight 16.7 per cent. Static deflection 2.66 in.) Curve I, 15 m.p.h.; II, 25 m.p.h.; III, 50 m.p.h.

Having thus obtained the displacement and velocity of the wheel when it clears the bump, we can apply equations (4) and (5) to obtain the subsequent motion until the wheel hits the road again, as was done in the case of "flat" irregularities. (The upward velocity attained by the body during the impact can be easily approximated.)

The foregoing analysis was based on the assumption that the wheel

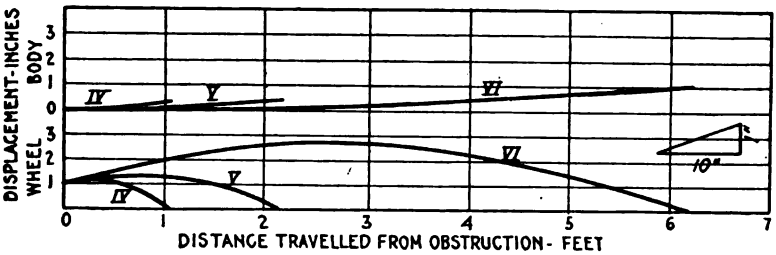

FIG. 9—WHEEL AND BODY TRAJECTORIES—EFFECT OF SPEED (Unsprung weight 25 per cent. Static deflection 2.66 in.) Curve IV, 15 m.p.h.; V, 25 m.p.h.; VI, 50 m.p.h.

rises a distance equal to the original deflection of the tire before the wheel center comes directly over the bump. It frequently happens, however, that this height is not attained until the bump is behind the wheel center. In this case it is sufficiently accurate to end Stage I when the wheel center comes directly over the bump, to skip Stage II altogether and let the final conditions of Stage I be the initial con-

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ditions of Stage III. That is, t_1 will be given by $t_1 = \frac{D}{V}$; and the final conditions of Stage I and the initial conditions of Stage III are obtained by substituting this in equation (11). The values of y_1 and v_1 thus obtained are then to be substituted in the

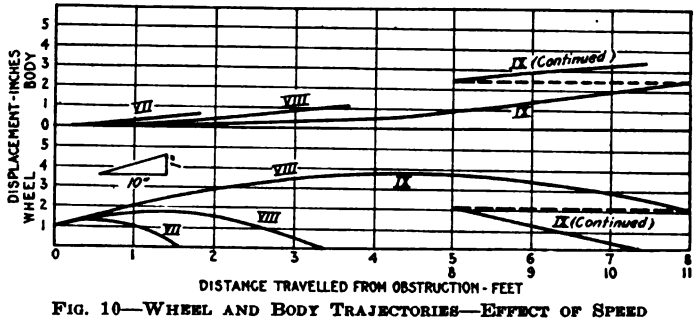

FIG. 10—WHEEL AND BODY TRAJECTORIES—EFFECT OF SPEED
(Unsprung weight, 41.7 per cent. Static deflection 2.66 in.) Curve VII, 15 m.p.h.; VIII, 25 m.p.h.; IX, 50 m.p.h.

equations for Stage III in place of y_2 and v_2. The only restriction which need be imposed in this case is that the bump shall not be too small, but very small bumps are of no practical importance.

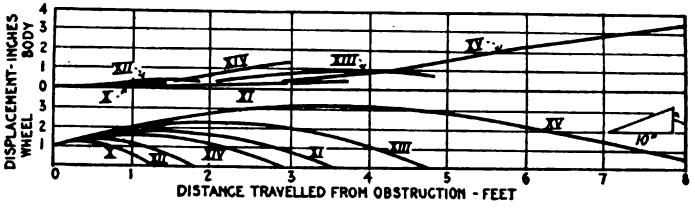

FIG. 11—WHEEL AND BODY TRAJECTORIES—EFFECT OF SPEED AND UNSPRUNG WEIGHT
(Static deflection 1.33 in. Unsprung weight 16.7 per cent for curves X and XI; 25 per cent for XII and XIII; 41.7 per cent for XIV and XV.) Curves X, XII and XIV, 25 m.p.h.; XI, XIII and XV, 50 m.p.h.

IV—WHEEL AND BODY TRAJECTORIES AND THEIR INTERPRETATION

A—*Data Relating to the Curves*

In order to illustrate the significance of the preceding analysis and to show its application to practical problems, I have plotted a number of trajectories of wheel and body, covering a fairly wide range of conditions. The principal factors are, for "flat" obstructions, the speed of the vehicle, the percentage of unsprung weight, the stiffness of the spring, and the dimensions of the obstruction; and for "sharp" irregularities, there are the additional variables wheel diameter, stiffness of the tire against road-bed, (static deflection of tire) and stiffness of tire against obstruction.

For purposes of plotting, the following data for "flat" irregulari-

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ties have been chosen: Speeds of 15, 25 and 50 m.p.h.; percentages of unsprung weight of 16.7, 25 and 41.7, the total weight in all

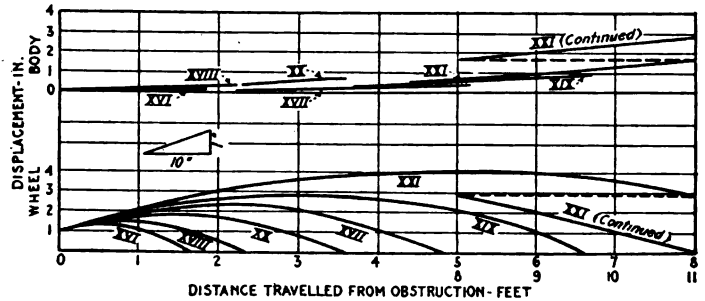

FIG. 12—WHEEL AND BODY TRAJECTORIES—EFFECT OF SPEED AND UNSPRUNG WEIGHT

(Static deflection 4 in. Unsprung weight 16.7 per cent for curves XVI and XVII; 25 per cent for XVIII and XIX; 41.7 per cent for XX and XXI.) Curves XVI, XVIII and XX, 25 m.p.h; XVII, XIX and XXI, 50 m.p.h.

cases being taken as 960 lb.; static spring deflections of 1.33, 2.66 and 4 in.; and two obstructions, one 1 in. high by 10 in. long, the other 2 in. high by 12 in. long. Of the fifty-four combinations which can be made of these variables, twenty-nine have been selected for plotting. Complete data for these are given in Table I. The last column in the table, namely, the upward velocity of wheel upon leaving obstruction, is calculated from equation (1) of the preceeding section.

For "sharp" obstructions the speeds chosen are 25 and 50 m.p.h.; the percentages of unsprung weight, 16.7 and 41.7, the total weight again being 960 lb.; static deflection of spring, 2.66 in.; stiffness of

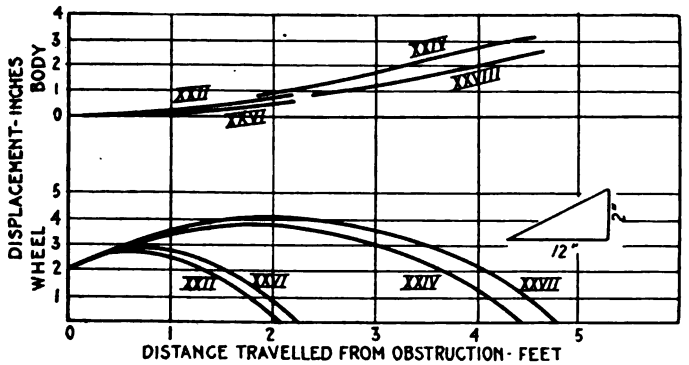

FIG. 13—WHEEL AND BODY TRAJECTORIES—EFFECT OF DEFLECTION AND UNSPRUNG WEIGHT

Static deflection, curves XXII and XXIV, 2.66 in.; XXVI and XXVIII, 4 in. Unsprung weight 16.7 per cent for XXII and XXVI, 41.7 per cent for XXIV and XXVIII. Speed 25 m.p.h.

tire against road-bed, 1920 lb. per in. (corresponding to a static deflection of the tire of ½ in.); stiffness of tire against obstruc-

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TABLE I—SPRING CHARACTERISTICS WITH "FLAT" OBSTRUCTIONS

Curve No.	Sprung Weight, Lb.	Unsprung Weight, Lb.	Unsprung Weight, per Cent	Velocity of Vehicle, M.P.H.	Stiffness of Spring, Lb. per In.	Height of Obstruction, In.	Length of Obstruction, In.	Normal Deflection of Spring, In.	Upward Velocity of Wheel Upon Leaving Obstruction, Ft. per Sec.
1	800	160	16.7	15	300	1	10	2.67	2.20
2	800	160	16.7	25	300	1	10	2.67	3.67
3	800	160	16.7	50	300	1	10	2.67	7.33
4	720	240	25.0	15	300	1	10	2.67	2.20
5	720	240	25.0	25	300	1	10	2.67	3.67
6	720	240	25.0	50	300	1	10	2.67	7.33
7	560	400	41.7	15	210	1	10	2.67	2.20
8	560	400	41.7	25	210	1	10	2.67	3.67
9	560	400	41.7	50	210	1	10	2.67	7.33
10	800	160	16.7	25	600	1	10	1.33	3.67
11	800	160	16.7	50	600	1	10	1.33	7.33
12	720	240	25.0	25	540	1	10	1.33	3.67
13	720	240	25.0	50	540	1	10	1.33	7.33
14	560	400	41.7	25	420	1	10	1.33	3.67
15	560	400	41.7	50	420	1	10	1.33	7.33
16	800	160	16.7	25	200	1	10	4.00	3.67
17	800	160	16.7	50	200	1	10	4.00	7.33
18	720	240	25.0	25	180	1	10	4.00	3.67
19	720	240	25.0	50	180	1	10	4.00	7.33
20	560	400	41.7	25	140	1	10	4.00	3.67

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21	560	400	41.7	50	140	1	10	4.00	7.33
22	800	160	16.7	25	300	2	12	2.67	6.12
23	800	160	16.7	50	300	2	12	2.67	12.2
24	560	400	41.7	25	210	2	12	2.67	6.12
25	560	400	41.7	50	210	2	12	2.67	12.2
26	800	160	16.7	25	200	2	12	4.00	6.12
27	800	160	16.7	50	200	2	12	4.00	12.2
28	560	400	41.7	25	140	2	12	4.00	6.12
29	560	400	41.7	50	140	2	12	4.00	

TABLE II—SPRING CHARACTERISTICS WITH "SHARP" OBSTRUCTIONS

Curve No.	Sprung Weight, Lb.	Unsprung Weight, Lb.	Percentage Unsprung Weight	Velocity of Vehicle, mi./hr.	Stiffness of Spring, lb./in.	Height of Obstruction, In.	Stiffness of Tire Against Road, lb./in.	Stiffness of Tire Against Obstruction, lb./in.	Normal Deflection of Spring, In.	Normal Deflection of Tire, In.	Wheel Diam., In.	Upward Velocity of Wheel Upon Leaving Obstruction, ft./sec.	Displacement of Wheel Upon Leaving Obstruction, In.
30	800	160	16.7	25	300	2	1920	1500	2.67	0.5	36	7.34	1.84
31	800	160	16.7	50	300	2	1920	1500	2.67	0.5	36	6.35	0.90
32	560	400	41.7	25	210	2	1920	1500	2.67	0.5	36	4.66	1.20
33	560	400	41.7	50	210	2	1920	1500	2.67	0.5	36	3.15	0.43
34	560	400	41.7	25	210	2	1920	1500	2.67	0.5	48	4.87	1.44

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tion, 1500 lb. per in.; height of obstruction 2 in.; and wheel diameters of 36 and 48 in. Of the possible combinations of these quantities, I have chosen five for plotting. Complete data for these are given in Table II. The last column, namely, upward velocity of wheel at the end of impact, is the v_0 of the preceding section.

The displacements of body and wheel from their normal positions are plotted as ordinates against distance traveled, and not against time, as abscissae. Inasmuch as only three velocities are employed in the curves, and frequently only two, it is a simple matter to pass to time as abscissae by dividing by the corresponding velocity.

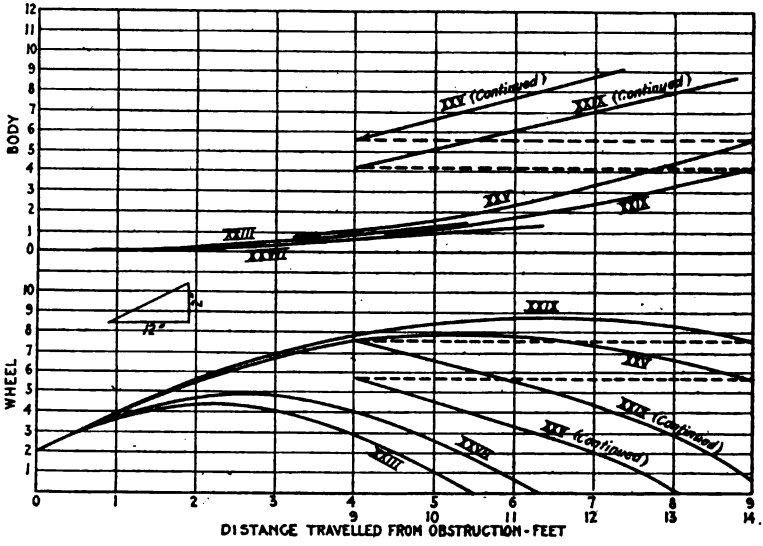

FIG. 14—WHEEL AND BODY TRAJECTORIES—EFFECT OF DEFLECTION AND UNSPRUNG WEIGHT

Static deflection, curves XXIII and XXV, 2.66 in.; XXVII and XXIX, 4 in. Unsprung weight 16.7 per cent for XXIII and XXVII, 41.7 per cent for XXV and XXIX. Speed 50 m.p.h.

The displacement scale is three times as large as the distance-traveled scale in all the trajectories except the last five, which are plotted together in Fig. 15 and have the displacement scale six times as large as the distance-traveled scale.

In order to investigate the effect of varying the percentage of unsprung weight, keeping the total weight constant, it is necessary to vary the spring stiffness so as to maintain the ratio of spring stiffness to sprung weight a constant; the only fair basis for comparison of different percentages of unsprung weight is a constant static deflection of the spring. This basis will be adhered to throughout.

The equations derived herein describe the motion of wheel and body up to the instant when the tread touches the ground again, so

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that the trajectories as plotted here end at this point. There is, of course, a secondary impact when the wheel hits the road-bed, but the effect of this will be small, in general, compared to that of the primary

TABLE III—MAXIMUM DEFLECTION OF SPRING AND MAXIMUM FORCE PER POUND IMPRESSED ON BODY—CALCULATED FROM THEORY

Curve No.	Max. Defl. of Spring, In.	Max. Force per Pound Impressed on Body
1	1.1	0.41
2	1.3	0.49
3	2.05	0.77
4	1.15	0.43
5	1.4	0.525
6	2.45	0.92
7	1.2	0.45
8	1.6	0.60
9	3.3	1.24
10	1.2	0.90
11	1.65	1.24
12	1.25	0.94
13	1.9	1.42
14	1.5	1.12
15	2.7	2.02
16	1.3	0.32
17	2.25	0.56
18	1.45	0.36
19	2.7	0.68
20	1.75	0.44
21	3.5	0.88
22	2.5	0.94
23	4.1	1.54
24	3.3	1.24
25	6.5	2.44
26	2.7	0.68
27	4.6	1.15
28	3.6	0.90
29	7.3	1.82
30	2.7	1.0
31	1.8	0.675
32	2.2	0.825
33	1.8	0.675
34	3.2	1.2

impact. The principal disturbance is over when the wheel has returned to earth; the motion of the body after this is mainly a decaying "free oscillation," of which we shall have more to say presently.

The maximum deflection of the spring, which is equal to the maximum difference between the wheel displacement and body displacement, is readily obtained in each case from the curves. Multiplying this by the corresponding stiffness of spring gives the maximum force impressed on the body due to the impact, and dividing this by the weight of body gives the maximum force impressed on the body

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per pound of body. The importance of this last quantity (which is numerically equal to one-thirty-second of the maximum acceleration of the body) requires no emphasis. The maximum deflections and forces per unit of weight have been computed in this way for each trajectory and the results are given in Table III. These deflections are due to impact, and do not include the static deflection of the spring. Some of these data are plotted in Figs. 16, 17 and 18.

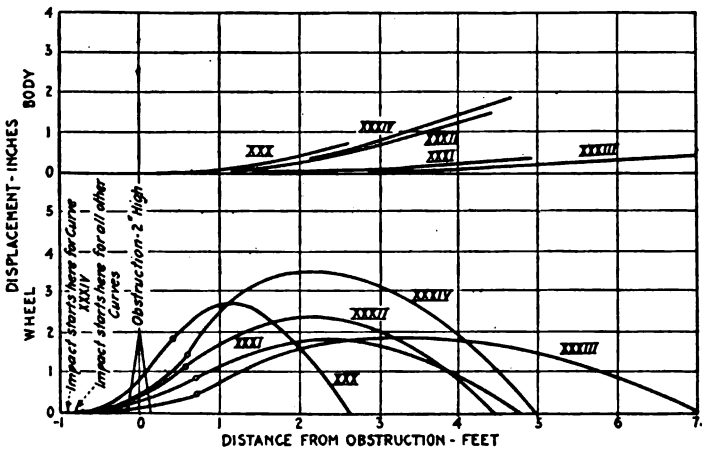

FIG. 15—WHEEL AND BODY TRAJECTORIES FOR "SHARP" BUMP

(Small circles mark points where impact ends.) Unsprung weight, 16.7 per cent for curves XXX and XXXI, 41.7 per cent for XXXII, XXXIII and XXXIV. Curves XXX, XXXII, XXXIV, 25 m.p.h.; XXXI and XXXIII, 50 m.p.h. Static deflection of spring, 2.66 in.; of tire ½ in. Stiffness of tire against obstruction, 1500 lb. per in. penetration. Wheel diameter, 48 in. for XXXIV, 36 in. for all others.

B—*Effect of Speed on Trajectories*

The effect of speed on the trajectories when the obstruction is "flat" appears very clearly in Figs. 8, 9 and 10. In Fig. 8, for example, the only difference between the curves is one of speed. The same is true of Fig. 9, the curves of which differ from Fig. 8 in having a larger percentage of unsprung weight. In Fig. 10 the unsprung weight is still larger, the data otherwise being the same as for Figs. 8 and 9. The effect of increasing speed is obviously to lengthen the wheel trajectory, and to heighten it, and to increase the displacement of the body. The length and height of the wheel trajectory, moreover, increase more rapidly than the first power of the speed, because the time of the flight is greater at greater speeds.

In a qualitative way we see that for a "flat" obstruction the effect of speed on the wheel trajectory is similar to the effect of muzzle velocity on the path of a projectile, the angle of elevation being constant.

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From Figs. 8, 9, 10, or more clearly from Figs. 11 and 12, we see that the effect of speed on the trajectories is greater, the greater the unsprung weight. It is instructive to note, for example, how much larger the body displacement is, and how much longer and higher the wheel trajectory is for Curve IX than for Curve III, the only difference being that the percentage of unsprung weight in IX is 2.5 times that in III.

The effect of speed on the maximum force per unit weight impressed on the body, as obtained from the first nine curves, is shown plotted in Fig. 16. It will be seen that the curves for the larger un-

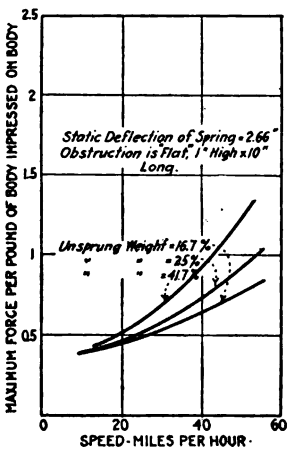

FIG. 16—MAXIMUM FORCE AS FUNCTION OF SPEED

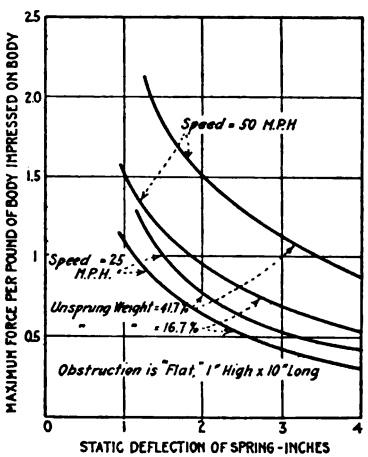

FIG. 17—MAXIMUM FORCE AS FUNCTION OF STATIC SPRING DEFLECTION

sprung weights are higher and steeper than for the smallest unsprung weight, showing again that the greater the unsprung weight, the greater are the effects of speed when the obstruction is "flat." Also, the fact that these curves are concave upward shows that the forces impressed on the body increase more rapidly than the first power of the speed, when the obstruction is "flat." Note that it is only in the extreme cases of the conditions specified in the first nine curves, that the force per unit weight impressed on the body exceeds unity, that is, the acceleration exceeds gravity.

In general, a study of all the trajectories for "flat" irregularities shows that the larger the irregularity (as we should expect) and the greater the unsprung weight, the worse are the effects of increasing speed.

Attention should be called to the fact that even for what we have called "flat" obstructions, the deformability of the tread cannot be altogether neglected. The influence of this deformability is to make

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the effects of increasing speed less marked, without, however, altering the general character of the curves.

When the obstruction is "sharp" and the tread highly deformable, the effect of increasing speed is *to lower the height of the wheel trajectory and to decrease the body displacement.* That is, for "sharp" bumps and highly deformable treads, the effect of speed is the reverse of that for "flat" bumps. This is shown by Curves XXX

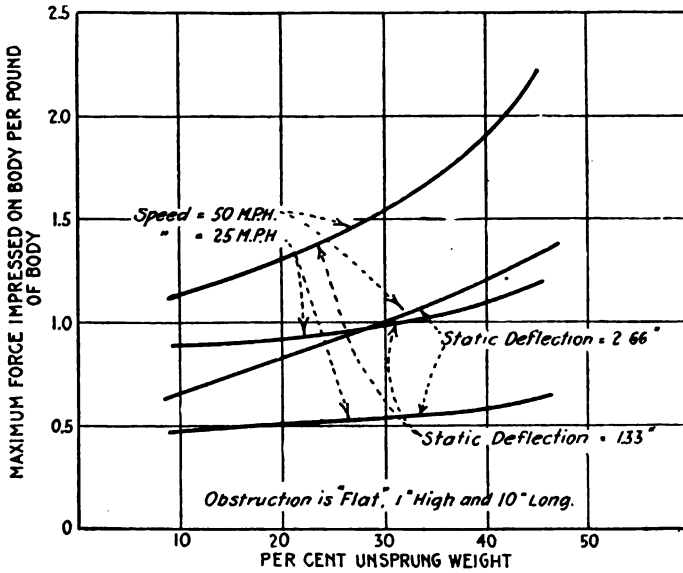

FIG. 18—MAXIMUM FORCE AS A FUNCTION OF WEIGHT DISTRIBUTION

and XXXI of Fig. 15, whose only difference is one of speed. The wheel trajectory is lower for the higher speed, and also longer, of course, but not in proportion to the increase of speed. Curves XXXII and XXXIII, whose only difference again is one of speed, but which differ from XXX and XXXI in having a larger unsprung weight, show the same thing as Curves XXX and XXXI. But here, it will be noticed, the effect of speed on the wheel trajectory is less than in the case of the smaller unsprung weight, while the influence on the body is substantially the same.

In Fig. 15 the obstruction is drawn in its proper place, and to scale. The distance from the peak of the bump to each curve is the penetration of the bump into the tire in each case. It is interesting to note the increase of this penetration with increase of speed.

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We see, then, that there are two opposing effects, one tending to improve the riding at higher speed, the other to make it worse. It will be observed, however, that the bad effects of increasing speed are much greater for "flat" bumps than the good effects for "sharp" bumps, so that the resulting tendency is usually for the riding to get worse with increasing speeds, although over certain types of roads it may actually get better. So far as forces impressed on the wheel and tire wear are concerned, however, they are always worse at higher speeds.

C—Effect of Weight Distribution

A glance at all our curves for "flat" irregularities, and particularly at Figs. 11 to 14, shows very strikingly the important part played by weight distribution. Keeping the ratio of spring stiffness to sprung weight constant, it is clear that, for flat bumps, the greater the percentage of unsprung weight, the higher and longer is the wheel trajectory and the greater the body displacement; in short, the worse is the riding. As a moderate example, note in Fig. 12 the difference between Curves XVI and XX, which differ only in unsprung weight; increasing the unsprung weight 2.5 times has increased the height of the wheel trajectory about 1.4 times, its length about 2.1 times and the body displacement at the end of the trajectory *about 5 times.* Note also the increasing importance of weight distribution with increasing speed and increasing size of obstruction. Comparing Curves XXVII and XXIX, for example, which differ only in weight distribution, we see that for the small unsprung weight the wheel trajectory attains a maximum height of about 4.8 in., and is about 6.3 ft. long, whereas for the large unsprung weight the maximum height is 8.7 in. and the length about 14.5 ft.; also, for the small unsprung weight, the body displacement at the end of the trajectory is about 1.3 in., whereas for the large unsprung weight *it is about 9 in.* These curves, it should be remembered, are for a 2-in. bump.

The manner in which the maximum force per unit of weight impressed on the body varies with the percentage of unsprung weight is shown, for the case of a 1-in. bump, in Fig. 18. Here again we see that the higher the speed, the more important is the unsprung weight. Note that for the most extreme conditions specified in Fig. 18, namely, small static deflection of spring, large unsprung weight and high speed, the maximum force per unit weight impressed on the body exceeds 2; the maximum force exceeds twice the weight of the body, even though the bump is only 1 in. high.

The unsprung weight exercises an important influence on the power required to drive the vehicle, and on the horizontal component of the blow on the wheel, which component, it should be noted, is felt directly by the body. The upward velocity of the wheel upon leaving an obstruction represents a certain amount of energy. This energy must come from the forward velocity of the vehicle as a

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whole, and the means by which the transfer of energy is accomplished is the horizontal component of the force on the wheel. It follows, therefore, that, for flat irregularities, the greater the unsprung weight, the more power required to drive the vehicle and the greater are the fore-and-aft-forces on the body.

What we have said so far about unsprung weight refers to "flat" obstructions. In practice the curves will be somewhat modified by the deformability of the tread, which cannot be altogether neglected even for "flat " bumps; but the general character of the results is not affected by this deformability.

The influence of weight distribution for the case of "sharp" obstructions is shown in Fig. 15. Comparing Curves XXX and XXXII, which differ only in weight distribution, we see that the larger unsprung weight has a *lower* but *longer* wheel trajectory, and a considerably greater tire indentation; also, the body displacement, while somewhat slower, is greater at the end of the wheel trajectory for the greater unsprung weight. Again comparing Curves XXXI and XXXIII, which differ from each other only in unsprung weight and from XXX and XXXII only in speed, we see that at the higher speed the two wheel trajectories are more nearly of the same height and the end of the body curve is only slightly higher for the larger unsprung weight than for the smaller. In all cases, however, for both kinds of bumps, the larger unsprung weight gives a longer trajectory. This has several disadvantages, as well be pointed out presently.

We see, then, that the undesirable effects of large unsprung weights on the riding qualities are obliterated, to a certain extent, in the case of "sharp" bumps, by the deformability of the tire, but *at the expense of the tire.* In fact, if we should plot a sufficient range of curves for "sharp" bumps, we should probably find certain conditions where large unsprung weights give better riding than small. This, however, would be achieved in a very limited number of cases only, and at the cost of increased tire wear. In general, we can say that the disadvantages of large unsprung weights far outweigh the minimizing influence of the deformability of the tire in the case of "sharp" obstructions, particularly in view of the undesirability of a long trajectory.

There is one possible benefit of a large unsprung weight, however, which may partly account for the widespread use of the heavy rear axle now in vogue. A large unsprung weight always has the tendency to reduce the upward velocity of the wheel upon clearing the obstruction; a tendency which is slight in the case of "flat" bumps, but very marked for "sharp" ones. Thus, from Table III, the v_o for Curve XXX is 7.35 ft. per sec., whereas $v_o = 4.66$ for Curve XXXII, which has 2.5 times the upsprung weight of XXX but is otherwise identical with it. Also for XXXI, $v_o = 6.35$, whereas for XXXIII, $v_o = 3.15$. This upward velocity, it is clear, is a measure of the rate, or suddenness, of flexure of the spring. Now, if our present springs have a serious im-

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perfection, and if this imperfection consists in an increasing resistance to flexure as the rate of flexure is increased, then large unsprung weights will give better riding than small, particularly over "sharp" bumps. This matter will be reverted to presently.

D—Effect of Spring Stiffness

In discussing the effects of varying the spring stiffness, it should be noted that it is only the *relative* stiffness, the ratio of sprung weight to spring stiffness, which is of importance. But this ratio is simply the static deflection of the spring due to the load it supports; hence, in what follows, we shall speak of static deflection instead of stiffness.

The influence of static deflection can be seen by comparison of Figs. 11 and 12, or more clearly from Figs. 13, 14 and 16. In Fig. 13, for example, Curves XXII and XXVI are identical except for static deflection, and likewise for the pairs XXIV-XXVIII, XXIII-XXVII, XXV-XXIX. These curves show that increasing the static deflection not only causes the expected lowering of the body trajectory, but also raises and lengthens the wheel trajectory. For this reason the improvement in riding resulting from an increase in static deflection is not proportional to this increase. This appears very clearly in Fig. 17, which shows how the maximum force per unit of weight impressed on the body varies with the static deflection. It will be observed that increasing the deflection from 1 to 2 in. reduces the maximum impressed force considerably more than increasing it from 2 to 3 in., and still more than increasing it from 3 to 4 in. In other words, *it is not so important to have a large deflection as it is to avoid a small one.* Daily experience brings this fact out very strikingly. With one passenger in a light touring car the riding may be disagreeable; addition of another passenger produces a marked improvement, of still another, a less marked improvement, etc. With motor trucks, where the live load is a bigger percentage of the sprung weight, this phenomenon is much more noticeable.

A large full-load deflection is desirable, therefore, in order to avoid too small a no-load deflection. Obviously it is also desirable to have the load variations as small as possible—to have as large a ratio of sprung dead-load to live-load as possible. It should also be remembered that the percentage of unsprung weight is much greater at light loads than at heavy loads.

From the foregoing, it is clearly advantageous to employ a suspension of increasing stiffness with increasing loads, a principle which is employed in some trucks but which has not met with widespread application.

E—Effect of Wheel Diameter

It is generally assumed that increasing the wheel diameter improves the riding qualities. I have not plotted enough curves to

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discuss this point completely, but in the one case which I did work out—Curve XXXIV of Fig. 15, which has the same data as Curve XXXII except for wheel diameter—the result of increasing the wheel diameter is seen to be an increase in the time of impact, a raising and lengthening of the wheel trajectory, and an increase in the body displacement. In short, increasing the wheel diameter has made the riding worse, in this particular case. Note, however, that the indentation of the bump into the tire is less. This is not to be taken as meaning that large wheels are undesirable. The use of large diameters has compensating advantages; namely, bridging over of small irregularities and reduction in the fore-and-aft component of the blow on the wheel. To illustrate, consider an extreme case: If a wheel of 16-in. diameter meets a 6 or 7-in. "sharp" bump, the result is practically a head-on collision; a 48-in. wheel, however, could probably surmount the obstruction without very great difficulty.

F—Disadvantages of a Long Trajectory

In order to drive and steer an automobile, it is necessary to maintain a certain average force between wheel and road. When there is no contact between wheel and road, this force is zero; hence, the longer the trajectory, other things being equal, the smaller are the average tractive and directive forces. This brings us to one of the most important advantages of a low unsprung weight—*the ability to hold the road.* In view of the foregoing discussion on the influence of unsprung weight and speed, it is unnecessary to enlarge upon this matter further than to point out that at higher speeds, just when greater tractive and directive forces are necessary, the time of flight of the wheel tends to become longer, thus reducing these forces and thereby making a low unsprung weight all the more necessary.

When the driving wheels leave the road, the resisting torque vanishes and the applied engine torque goes into increasing the rate of revolution. This increase in speed of the driving wheels, and the energy it represents, are greater, the longer the trajectory, and, of course, are given up when the wheel comes in contact with the road again. Hence another disadvantage of the long trajectories due to large unsprung weights is the greater slipping of the driving wheels and corresponding increased wear on the tires, and also the greater fuel consumption necessitated thereby.

The above remarks, it should be noted, apply with much less force to "sharp" irregularities than to "flat" ones.

V—FRICTION IN THE SUSPENSION

A—Effect of a Series of Bumps

In all of the preceding discussion the wheel and body were supposed to be in vertical equilibrium upon encountering the obstruction.

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In operation, however, irregularities usually come in rapid succession; when a fresh bump is encountered, the disturbance due to those immediately preceding has not, in general, subsided. This disturbance may make the effects of the new impact more severe, or less so, depending upon circumstances. The most important case is where the successive obstructions are encountered at such intervals as to superimpose their effects in a cumulative way. This case is of considerable practical importance and will be discussed in some detail.

Imagine a weight hung from a fixed support by means of a spring and free to oscillate vertically. If the weight is given a displacement or a blow and then left to itself, it will execute vibrations of a definite period, which are known as *free vibrations.* In particular, because of frictional losses, the vibrations will die down with more or less rapidity, and we have a *decaying free vibration.* But now suppose that the support, instead of being fixed, is given a periodic vertical motion; this impresses a force of that period on the body, which will ultimately settle down into a steady vibratory motion also of that period. These are known as *forced vibrations.* If the period of the forced vibrations should happen to coincide, or nearly so, with that of the free vibrations, the well known phenomenon of resonance steps in and the amplitude of the resulting oscillation may become extremely large. *The only force tending to limit this amplitude is the friction in the spring and mountings.*

The above remarks may be applied word for word to a vehicle in motion. If disturbances are encountered at a frequency coinciding more or less closely with the frequency of the free vibrations of the body, the latter will tend to be set into violent oscillation. To guard against this contingency, a certain amount of friction in the suspension is absolutely imperative, as friction is the only force limiting these oscillations, except, of course, breakage of the spring, "hitting bottom," or some other form of stop.

On the other hand, friction in the suspension reduces the freedom of relative motion between wheel and body, and so tends to increase the *forced vibrations.* The practical problem is, then, to introduce enough friction to damp out the free vibrations very quickly, without unduly increasing the forced vibrations.

The best solution of this problem is to make use of the fact that free vibrations of the body are unobjectionable *if the amplitude is small.* That is, we may allow the relative movement between body and axle to be as free as possible for small displacement and begin to introduce friction when the relative displacement exceeds a certain amount, increasing the friction as the displacement increases.

This method, which is employed in some recent "shock-absorbers," is by no means the only one. As a matter of fact, the inter-leaf friction of ordinary springs is usually sufficient for most cases; it is of a desirable kind, too, for it tends to increase with increasing

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deflection. In some cases additional friction in the suspension is certainly desirable; for example, where the automobile tends to "pitch" in going over certain roads. On the other hand, the present tendency to lubricate spring leaves is in the opposite direction; and certainly it is very contradictory practice to lubricate the leaves and then add a sliding surface frictional shock-absorber, for example.

"Taking care of the rebound," which is the avowed function of most frictional shock-absorbers, is, it should be noted, simply reducing the violence of the first swing of the free vibration following an extreme displacement.

B—Surface Friction and Its Effect on the Trajectories

The friction necessary to damp out the free oscillations may be applied in different ways and follows different laws; for instance, it may be constant, as in one type of frictional shock-absorber; it may increase with increasing displacement, as already mentioned; it may be intermediate between these two, as in the inter-leaf friction* of the springs themselves; or it may be proportional to the *rate of flexure*, as in the case of oil friction. The last-named will be considered separately. The present remarks apply to sliding surfaces or "static friction" only.

The actual value of the friction in leaf springs is very variable. As a rough average we can take the frictional force as 15 to 30 per cent of the elastic force. This means that static friction of the order of magnitude of one-fourth or so of the elastic force is generally sufficient to keep the free oscillations from becoming excessive, although at times more friction would be desirable. On the other hand, this means that the maximum force per unit of weight impressed on the body, such as given in Table III and plotted in Figures 16, 17 and 18, will be also increased 15 to 30 per cent, assuming that no other frictional forces are present.

During the impact and the first part of the trajectory, the wheel is moving toward the body; hence the friction of the suspension will raise the body curve during this period and lower slightly the wheel curve. Just before the wheel trajectory reaches its maximum height, the wheel and body begin to separate from each other; hence the friction will then tend to lower the body curve and raise the wheel curve. The most important effect of friction, however, is the increase it produces in the forces impressed on the body. If this increase is not more than 50 per cent., let us say, it cannot be regarded as highly objectionable. But if it should reach several hundred per cent., or more, as it does in at least one case of oil friction, the price paid for minimizing the free oscillations is altogether too high.

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*Influences Affecting the Fundamental Deflection of Leaf Springs, David Landau, S. A. E. TRANSACTIONS, 1914, Part I, p. 305.

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C—Oil Friction

There is on the market at the present time an air spring, a piston and cylinder arrangement in which oil is used for two purposes—to prevent air leakage and to damp the free oscillations. From such crude tests as setting the body in vibration by muscular force and noticing the damping upon removing the force, I find that the spring has more than enough friction to render the suspended system aperiodic; that is, the free oscillations die out in one swing. Also, the friction is due almost entirely to the oil. Oil friction, the force resisting the flow of oil through an orifice, increases rapidly with increase of rate of flow. We can assume that this friction is proportional to the first power of the velocity. It is known from the theory of vibrations that if a simple vibratory system has friction proportional to the velocity, the system will be aperiodic if

$$R \gtreqless 2\sqrt{MK}$$

where R is the frictional force per unit of velocity, M the mass of the body and K the stiffness of the spring. Let us suppose that we have this type of friction in a definite case, and apply this formula to it, for example, Case V in Table I. We get

$$R \gtreqless 2\sqrt{\frac{720}{32} \times 270 \times 12} \text{ or } R \gtreqless 540$$

That is, the frictional force per unit of velocity is equal to or greater than 540 lb. per ft. per sec. Now the maximum velocity of approach of wheel toward body is, in Case V, about 3.6 ft. per sec.; hence, the maximum frictional force impressed on the body is equal to or greater than 540 $\times$ 3.6, or about 2,000 lb. The corresponding maximum elastic force impressed on the body is, from Table III, 0.525 $\times$ 720, or about 380 lb. In this particular case, therefore, which, by the way, is a mild example, the frictional force is more than five times the pure elastic force. If we went through our list of examples, we should doubtless find conditions in which the frictional force exceeds ten or more times the elastic force.

The results of this calculation may seem preposterous, but we can feel absolutely certain that they are of the right order of magnitude. The formula $R \gtreqless 2\sqrt{MK}$ is easily verified, and that the suspension referred to is strictly aperiodic and the friction thereof proportional to at least the first power of the velocity is also easily verified. There can be no doubt, therefore, that the value of the frictional forces for the air-spring referred to are of the right order of magnitude.

We see then, that, if there is viscous friction in the suspension, and if the damping of the free oscillations is due entirely to this friction, forces will be impressed on the body far in excess of the

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normal elastic forces. There is, however, one oil dash-pot shock-absorber on the market which has a valve arrangement whereby the friction on the upward stroke is mostly eliminated. This device therefore acts only "on the rebound," but it has the additional feature that it retards the return of the wheels to the road, that is, it lengthens the trajectory, which has been shown to be disadvantageous. The best plan is to eliminate oil friction altogether and obtain the necessary damping by means of sliding surface friction.

VI—PRACTICAL APPLICATIONS OF THEORY—IS STEEL ELASTIC?

In the preceding sections, the actions which take place when different types of irregularities are encountered have been quantitatively described, on the assumption that only purely elastic forces exist in the suspension; the modifying influence of various kinds of friction has been discussed. The most important conclusions arrived at are: (1) the unsprung weight should be as small as possible; (2) a small static deflection at any load should be avoided; (3) there should be just enough friction in the suspension to quickly damp out the free oscillations of the body, and this friction should be independent of the rate of spring flexure; (4) except for this friction, only purely elastic forces should exist in the suspension.

Although these conclusions have been recognized by engineers in particular instances, there has been, in general, considerable difference of opinion about them. The chief value of the mathematical analysis lies in that it proves these results definitely; that it gives quantitative ideas as to how much gain is to be expected by a given increase in spring flexibility or a given reduction in unsprung weight; and that it furnishes an estimate of the magnitude of the forces impressed on the body in particular cases.

We may say, for example, that the substitution of wire wheels for wood wheels, while still retaining the live rear axle, would have a negligible effect on the riding qualities. Furthermore, cantilever springs offer a considerable reduction in unsprung weight, but even here the weight of the live rear axle renders such reduction of minor importance. On the other hand, the use of cantilever springs in conjunction with the double side chain drive or the DeDion drive, for example, would bring the unsprung weight almost as low as we can hope to reach, and should result in a very appreciable improvement in riding qualities.

Again, the theory shows that the use of a suspension whose stiffness increases with increasing load is more advantageous than is generally recognized. Suspensions of this kind have been worked out and are in use, but, as already mentioned, they have not met with widespread application.

But although these matters relating to spring stiffness and weight distribution are admittedly of considerable importance, daily

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experience with motor vehicles raises a more vital question—*Do only purely elastic forces, except for friction between the leaves and in the shackles, exist in present springs?* This question is more a matter to be settled by experiment than by theory, but no discussion of suspension dynamics can be complete without some reference to it.

One of the most striking features of the body trajectories in the previous sections is the absence of sharp displacements. A glance at Table III shows that only rarely does the maximum force impressed on the body exceed the weight of the body. In fact, the most violent force encountered in the examples is not much more than twice the weight of the body. Such forces would hardly be described as *blows*. On the other hand, in daily experience we habitually meet with forces which we would unhesitatingly class as *blows*, particularly in riding without pneumatics. (I am not referring here to "hitting bottom," or to forces transmitted to the driver through the steering gear.) The question arises: *Where do these forces come from?*

Viewing the matter from a different angle, we may call attention to the frequency with which air has been proposed as a substitute for steel as a suspension medium. Air, however, cannot be more than perfectly elastic, and if steel is approximately perfectly elastic, no advantage can accrue from substituting air for it. But many air springs have been built, which, although they may not appeal to engineers as a practical expedient, nevertheless throw a good deal of light on this question. I have had personal experience with a solid rubber-tired car whose body was mounted on air cushions, the damping forces coming from the friction in the mountings and in the fabric of the cushions themselves. I can say without reserve that this car rode markedly better than any other solid-rubber-tired vehicle in which I have ridden; that "hard" quality which generally characterizes riding without pneumatics was entirely absent. And I may add that other engineers who have had experience with air springs devoid of undue friction, subscribe to the same opinion.

AUTHOR'S CONCLUSION

There can be no doubt, therefore, that there is something peculiar about the behavior of present springs under rapid strain. A theoretical investigation which I hope to publish soon, shows that this peculiarity does not lie in the inertia of the spring, and that we must therefore look for it in the material of the spring. But steel under rapid strain has been the subject of many investigations in scientific laboratories, and the general results of these experiments tend to show that *steel is approximately perfectly elastic even under shock stresses*, just as it is under slow stresses, within the elastic limit.

There is, therefore, a wide discrepancy between laboratory experiments and road experience. In order to throw more light on this important question, I began experiments in the Hartley Research

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Laboratory of Columbia University some eighteen months ago. In a future paper I hope to give a complete description of these experiments, which have been extended to road tests, and which, it may be added, have gone a long way toward clearing up the discrepancy. It is to be reiterated at this time that the object of the present paper is to show quantitatively what happens when an irregularity is encountered under certain simple conditions, and to give a clear understanding of the fundamental laws involved in suspension dynamics.

DISCUSSION

M. H. LANDIS:—Spring suspension should be designed with two main points in view: First, to make the unsprung weight of cars a minimum; second, to provide flexible springs with a large range of movement and a minimum of inter-leaf friction. The curves given in Figs. 8, 9 and 10 of Dr. Liebowitz' paper show clearly that the forced vibrations are reduced to a minimum; that is, the body displacement is reduced the smaller the percentage of unsprung weight.

The less the inter-leaf friction, the less the coefficient of spring stiffness, reducing the impact or blow given to the body during the spring compression—that is, the greater are the shock-absorbing qualities of the spring. It is desirable to reduce the coefficient of spring stiffness, but any means employed to introduce a resistance to spring compression increases this coefficient. Shock absorbers are made that provide a resistance to the compression of the spring with the avowed purpose of damping its free vibrations. The greater the blow or impact, however, the greater the friction introduced in compression and the larger the coefficient of spring stiffness.

The ideal spring suspension should have such resilient qualities that it will absorb the slightest shocks and be of such strength and range of movement that it will absorb the maximum shocks. The spring should be of sufficient range to provide for variation in the live loads from the maximum to the minimum. A car will then ride as well with a light load as with a heavy load, regardless of the speed. To secure this result, we should increase the width of the spring leaves to obtain strength and reduce the number of leaves to minimize inter-leaf friction. The ideal spring suspension should have all inter-leaf friction eliminated and have a coefficient of spring stiffness determined by the physical properties of the steel.

It is sometimes said that the free vibrations of such a spring would be of such great magnitude that when the forced vibrations acted in harmony with the free vibrations the cumulative effects would make the spring unmanageable. This is true, but a means can be provided for the control of the free vibrations without interfering with the spring compression.

The evolution that gave us the present-day automobile necessitated a change in our designs of spring suspension. The change was made at the expense of easy-riding qualities. In the earlier days light

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weights and low speeds permitted flexible springs; higher speeds meant larger engines, heavier cars and demanded relief from springs "hitting bottom"—enter the friction shock absorber. Then came the stiffer springs to care for the heavier weights and higher speeds—comfort of the old days still lacking—enter the auxiliary-spring shock absorber, deeper upholstery and larger tires. The discomfort of the passenger was relieved to a great extent by the deeper upholstery. The large tires lessened both the effect of sharp bumps and the shock to the unsprung portion of the car.

The introduction of inter-leaf friction to dampen the free vibrations of the spring is not the correct solution of the problem. This friction increases the coefficient of stiffness of the spring; moreover it is variable, depending on the lubrication of springs. The inter-leaf friction to be really effective should become active only during spring recoil and be proportional to the rate of flexure. Tests show this resistance is practically negligible during spring expansion.

CONTROL OF RECOIL NECESSARY

The same types of springs employed on pioneer automobiles will fill the requirements of the present day. The flexibility of the early car is desired for comfort, safety and economy, but to obtain it we must provide a means for controlling the violent spring recoils without increasing resistance to spring compression and without placing an additional tension on the springs. This element must be active at all times, regardless of the live load of the car.

We tested the theory I have stated with a specially built car. The suspension was made so flexible that the car could be put in vibration with one hand by pressing on the radiator cap. The springs were made with a large range of movement; there were but three leaves to each member, with width of spring-leaf to compensate for the small number.

When this car was driven over average roads at any speed the springs were "hitting bottom" frequently; the conclusion naturally being that the springs were too flexible. When we provided the proper time element for controlling the recoil, "hitting bottom" ceased, regardless of the speed traveled. This was due to the fact that the effect of the free and forced vibrations was not cumulative. With the car traveling over choppy and uneven roads at speeds up to 50 m.p.h., the "bobbing" of the front end ceased.

We strongly recommend the use of front springs having a large range of movement. The front wheels meet the obstruction first. If the range of movement is sufficiently great and the spring is flexible, the impressed force tending to throw the car out of its straight line is small; the spring will never "hit bottom" so long as the height of bump is less than the range of its movement. The average front spring as now used is stiff and has but a small range of movement. The result is an increased impressed force tending to increase the body displacement. If the shock is excessive the spring "hits bottom"

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and the car is thrown bodily into the air. This remedy to overcome that "bobbing action" certainly fails to accomplish results.

Dr. Liebowitz states (page 210) that it is improper to retard the return of the wheels to the road as it lengthens the wheel trajectory. We fail to see the disadvantage of the increase of wheel trajectory. The wheel strikes the road while the spring is expanding with the free vibrations under control. The blow due to impact on the obstruction is met by a spring expanding with a force opposite to that due to the impact of the wheel with the road. The free vibrations and forced vibrations cannot act in harmony and make the springs "hit bottom."

HENRY HESS:—Will Dr. Liebowitz give some information on the

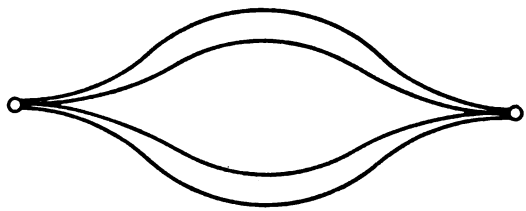

FIG. 19—"DOUBLE-SWEEP" SPRING CONSTRUCTION

increasing-stiffness type of spring that he refers to as used on trucks, and will he also sketch the characteristics of such a spring that his experience has shown to be desirable? I should like further to ask whether he has had experience in quickly damping vibrations and so holding down the summation of throws due to successive obstructions; in other words, preventing resonance summation without resorting to friction.

Mr. Landis finds that a flexible spring, built up of only three leaves, contrary to usual experience and practice, gives desirable and easy riding qualities. The single leaf spring can be so designed and mounted that it responds readily to slight obstructions, while opposing heavier or sudden obstructions with increasing stiffness. On slight obstructions the effect is that of an exceedingly easy-riding spring, while full benefit is had of a heavy and strongly resistant spring when it is necessary to resist greater forces. Such a spring can be so mounted that it is entirely aperiodic, and its vibrations can be confined almost entirely to the first throw, with succeeding vibrations so small that successive impulses due to successive obstruction cannot result in any summation of such impulses. The throw of such a spring is practically never greater than that due to any one single impulse.

DR. BENJ. LIEBOWITZ:—The Westinghouse air-spring theoretically gives a desirable characteristic that begins with a flexible spring and finishes with a spring of infinite stiffness, although it has the unfortunate oil friction that I mentioned. But many steel springs have been designed with similar characteristic. For instance, one type—I think it is called the "double-sweep" full-elliptic—is shown in Fig.

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19. As the spring is compressed, the effective length is shortened, so that the stiffness increases with the deflection.

HENRY HESS:—How would it act on its rebound?

DR. BENJ. LIEBOWITZ:—Much the same as an ordinary spring, except that the throw might be more violent if the rebound were unchecked. Again, a spring is used on the Marmon car that obtains the same effect by a cam. As the springs compress their effective length is shortened by progressively contacting with the cam. I suppose many springs have the characteristic of increasing stiffness with increasing load. The one having the characteristic shown in Fig. 20 is well known. Furthermore, a spring that comes into action

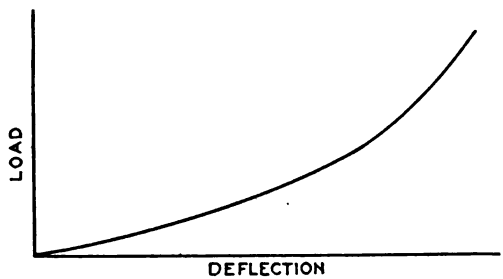

FIG. 20—CHARACTERISTIC OF SPRING IN WHICH STIFFNESS INCREASES WITH LOAD

only after the load is increased to a certain amount, is employed on many trucks. Such suspensions have a characteristic shown in Fig. 21, which for practical purposes is almost as good as the device in Fig. 20.

W. H. BROWN:—It is a law of mechanics that impending friction is greater than sliding friction, as evidenced in the skidding of the automobile. The same law applies to the spring. Another law is that the stress due to a load suddenly applied is double the stress when the load is gradually applied. The inter-leaf friction should be reduced at the instant the shock is transmitted to the spring. If it is reduced by eliminating rust, we will have suspensions that are better and that will give the same results under any conditions.

OPERATION OF SINGLE-LEAF SPRING

HENRY HESS:—A single-leaf spring is more than a possibility. I have made such a spring that gave me the characteristics of a flexible spring at the start and of a stiff spring at the end of the vibration. The single-leaf spring does away with the complications of inter-leaf friction present with the multiple-leaf type, with variable and uncertain friction and with shock absorbers. I have tested such springs and various others with an apparatus that traced the characteristics and periods automatically under a wide range of conditions. The springs were of uniform cross-section, though the section can also be variable. I did not support the spring at the center, but placed

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underneath a cam that made increasing contact along the length of the spring as it deflected under load; this caused a progressively increasing shortening of the spring and with that a progressively increasing stiffening with increasing deflection. The rate of vibration is a function of spring length; the length is continually changing and therefore rate of vibration also; the spring is accordingly damped after its first pulsations. The single-leaf spring, used without the cam, has the usual straight-line characteristic of a flexible spring; yet with the cam to support it has all the ultimate stiffness of a heavy multiple spring. With such construction the friction is eliminated and no lubrication is required.

VICE-PRESIDENT FOLJAMBE:—According to the paper it is not good

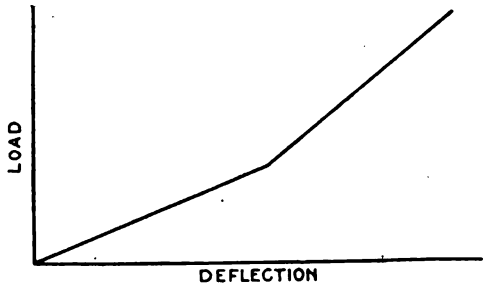

FIG. 11—CHARACTERISTIC OF SPRING THAT COMES INTO ACTION ONLY WHEN LOAD IS INCREASED TO A CERTAIN AMOUNT

practice to oil the springs. In actual practice springs are oiled more and more every day.

M. H. LANDIS:—It is absolutely necessary to lubricate the springs. If the attempt is made to control the free vibrations by inter-leaf friction, variable results will be obtained. The only way to eliminate this friction is to lubricate thoroughly. With inter-leaf friction reduced to a minimum we can design springs considering only the physical properties of the steel.

W. H. BROWN:—I advocate inter-leaf oiling. We built an instrument to record graphically the shock transmitted to the car body and found that if the springs are flexible on account of the elimination of friction between the leaves, many of the minor blows are entirely eliminated. In making a test run with a car with rusty springs, we obtained a graph much like the teeth of a large lumberman's saw. Using the same car on the same run with the springs freed from rust and oiled, we obtained a graph from which the large teeth had practically disappeared. The minor blows, which are the more frequent, are practically eliminated if the inter-leaf friction is kept low by a film of oil.

EFFECT OF TORQUE-ARM

A. C. WOODBURY:—This paper relates to "Vehicle Suspensions," and it is characteristic of the ordinary consideration of vehicle suspension that only the springs are mentioned. The effect of the

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torque-arm, or whatever takes the torque of the rear axle, is generally assumed to be negligible, although it has an important bearing on the riding of the car and on the stresses in transmission parts. Let us assume that the torque-arm is 42 in. long, which probably is correct for a car of small or medium size, and that the road wheel goes over a "flat" obstruction 2 in. high and 1 ft. long, of such form that the upward acceleration of the wheel is constant. We assume that an isosceles triangle with one side 2 in. and the other two sides 42 in. long is formed by the torque-arm in its two positions. The acute angle of that triangle would measure 2.6 deg. At a moderate car speed of 50 ft. per sec., which is about 35 m.p.h., the average angular velocity of the torque-arm would be 130 deg. per sec. As the obstruction was assumed to be such that the acceleration is constant, the final velocity is twice 130, or 260 deg. per sec. The arm acquires its velocity in one-fiftieth of a second; dividing 260 by 1/50 gives 13,000 deg. per sec. per sec. as acceleration; dividing that by 360 gives the acceleration as 36 rev. per sec. per sec. Since rotation of the rear-axle housing, while the propeller-shaft is prevented from rotating on its bearing, causes motion of the car, the acceleration of the torque-arm must result either in acceleration of the car (assuming constant engine-speed) or retardation of the engine (assuming constant car-speed).

It should be understood that both slippage of the tire and retardation due to increased road resistance are disregarded. A 4-to-1 gear ratio would give a retardation of the engine of 144 rev. per sec. per sec.

To give an idea of the magnitude of that retardation, assume an engine of such qualities that by pushing down the accelerator pedal it could be given that much acceleration instead of retardation. If the throttle were held down for one minute the speed of the engine would be 520,000 r.p.m. As a matter of fact the tire does slip, the car does accelerate, and the engine does decelerate, thus distributing the effects.

Assume that the engine does not retard but that the car accelerates. With a 32-in. wheel this acceleration is about 300 ft. per sec. per sec.; the acceleration of gravity is only 32.16 ft. per sec. per sec., which explains why a strong pressure in the back is felt in going over a bump. This effect is bound to occur in any car in which the rear-axle torque reaction is taken in such a way that a movement of the springs produces rotation of the axle. It occurs in a chain-driven car if the driving and driven sprockets are not of the same size, but is absent in the Hotchkiss drive and in some special constructions, such as that on the Lanchester car.

AUTHOR'S CLOSURE

DR. BENJ. LIEBOWITZ:—Mr. Woodbury's point is well taken, and it is a matter that ought to be investigated.

It is not possible to take care of free oscillations without friction. Free oscillations become dangerous when those due to an initial disturbance are successively reinforced by subsequent disturbances.

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Since road irregularities occur at random, this reinforcement of oscillations is just as apt to take place with one type of suspension as with another. To use a spring whose stiffness varies with the deflection, so that the periods of vibration of the car body are not well defined, is in itself no safeguard against this type of resonance phenomenon.

Free oscillations, once they are started, represent a certain amount of energy stored in the system. This energy must be dissipated, since we cannot convert it into a useful form, and hence friction in the suspension is absolutely necessary. No spring-cam arrangement and no adjustment of periodicities will enable us to get rid of free oscillations; under certain critical conditions they will always appear, unless a sufficient amount of friction is present in the suspension.

It would be desirable to introduce the friction on the rebound only, so as to have the spring free on the up-stroke. But for the same rate of damping, twice the amount of friction would be required as would be necessary if the friction acted both ways. Friction only on the rebound therefore has the disadvantage of retarding the return of the wheels to the road, thus lengthening the trajectory, which is undesirable.

Mr. Brown stated that the stress due to a suddenly applied load is twice that due to the same load gradually applied. While this is true, it must be correctly interpreted. It does not mean, for example, that if a weight is dropped from a height onto a floor, it will produce a stress in the supporting structure equal to only twice the weight. In all cases when the rule is applicable, the double stress is due to the oscillation produced by the sudden application of the load. The effects of such oscillations on springs have been considered in the paper. As Mr. Brown states, static friction is greater before than after it is overcome. But in leaf springs the rubbing of the leaf surfaces is so continual that this phenomenon can hardly enter to any appreciable extent.

In order to show that the inter-leaf friction is not what is causing the trouble, I compared an ordinary leaf spring directly with another spring whose riding qualities could safely be assumed as practically ideal. The ordinary leaf spring with its inter-leaf friction gave substantially as good riding as the practically ideal spring. This result is contrary to what we should expect from road experience, but agrees with the results previously obtained in scientific laboratories.

The experiments were continued and it was found that the ordinary spring did not maintain its fine riding properties, but after a while developed a markedly hard riding quality. According to this result, which I may add, is corroborated by daily experience rightly interpreted, the trouble with present springs is that after a comparatively short time in service they develop a type of fatigue, which is analogous to and doubtlessly connected with the well-known fatigue causing rupture. This hitherto unsuspected fatigue phenomenon is responsible for the hard riding qualities that characterize most of our motor vehicles.

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RECENT AEROPLANE-ENGINE DEVELOPMENTS

By Neil MacCoull
(Non-Member)

Abstract

The author gives a brief review of developments during the past year in the construction of aeroplanes, particularly as affected by the European War. He takes as an example the Renault twelve-cylinder engine, citing the respects in which the present differs from previous models. Such factors as the changes in cooling systems, method of drive, valve construction and starting devices are considered.

The requirements of aeroplane engines, such as constant service, high speeds (of aeroplanes) and stream-line form of engines and radiators, are outlined. Propeller requirements are dealt with at length, curves being given by which the efficiency and diameter of the propeller can be obtained. In conclusion a number of different engine installations are illustrated and compared.

The Great War is giving aviation a literal trial by fire which is proving it to be even more dependable and indispensable than the wildest dreamers had predicted; and as the fit are continually being separated from the unfit, progress is being steadily recorded. The progress during the year since the author presented a paper on Aeroplane Engines before this Society, has not been marked by the creation of radical designs, but by the detail refinement of engines involving the fundamentals of conventional automobile practice. Most aeroplanes are now expected to make long flights over water and hostile ground where engine failure often results in the death of the aviators as well as the loss of valuable military information. Reliability is of far greater importance than in times of peace, when an aviator can almost invariably glide to earth in safety with a dead engine.

It is because of the prime importance of reliability that the newest engines are all water-cooled; several makers are using some device to cool also the oil. Ignition must have been a frequent source of trouble at the beginning of the war, because it is now the rule to have two independent ignition systems; some of the twelve-cylinder engines are provided with four six-cylinder magnetos. The increased power resulting from the use of double ignition is not alone enough to justify the added complication and weight of double magnetos, except on engines running at the highest speeds.

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CHANGES IN RENAULT ENGINE

As an example of recent development, it is interesting to note in what respects the new twelve-cylinder Renault engine differs from previous models. The cooling system has been changed from forced air draft to water; the geared propeller drive is dispensed with and the engine speed lowered from 1800 to 1200 r.p.m.; magnetos and carbureters are increased from two to four; valves from F-head with one central camshaft in the V between cylinders to overhead with the camshaft over each row of cylinders, Mercedes fashion. The connecting-rod construction in which one rod is pinned to the side of the large end of the opposite rod must have proved satisfactory in the air-cooled types because it is retained in the latest type.

The recent adoption of the overhead valve with overhead camshafts by so many manufacturers suggests that valves are still a source of trouble and that other types do not stand up as well. The most popular construction now used is still far from ideal since the valves seat directly in the cylinders, which must be removed in order to change or regrind the valves, a not infrequent occurrence with high-duty engines. It is to be hoped that some of the rotary valves now being developed, and which look so promising on paper, will prove equally satisfactory in reality.

MUFFLERS AND MECHANICAL STARTERS

Mufflers and compressed-air starters are now considered as standard equipment on most engines. Mufflers have always been desirable in order to relieve the aviators of the nervous strain resulting from the terrific roar of high-power engines, and to make possible wireless receiving as well as conversation; but in the past many pilots preferred to listen to the exhaust, as it was a fairly good indicator of the way the engine was running. Now, however, mufflers are a necessity in many circumstances, such as raids or scouting trips over peaceful enemy country, where the roar of an engine would increase the possibility of detection.

Mechanical starters are invaluable in assisting a rapid get-away, whether on friendly or on enemy territory. They are generally operated by compressed air, and consist of a valve mechanism that distributes a compressed mixture directly to the cylinders, as do the starters of large stationary engines; or of a separate compressed air motor that cranks the engine, just as the electric starting motor does.

DETERMINATION OF SERVICE REQUIREMENTS

One difficulty frequently confronting aeroplane-engine designers when they are not familiar with aeronautics is the determination of the service requirements in order to strike a balance between weight of engine and fuel consumption, propeller speed, and installation requirements. There has been considerable difference of opinion in regard to these points even among aeronautic engineers, but they

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are now fairly well defined. An automobile engine must be designed to meet a large number of varying requirements such as rapid acceleration, good pulling power at both high and low speeds and flexibility. The service required of an aeroplane engine on the other hand is constant, it resembling the marine engine in this respect. To a certain extent this simplifies the question of design, since it allows of specialization for one particular class of service. The development of avi-

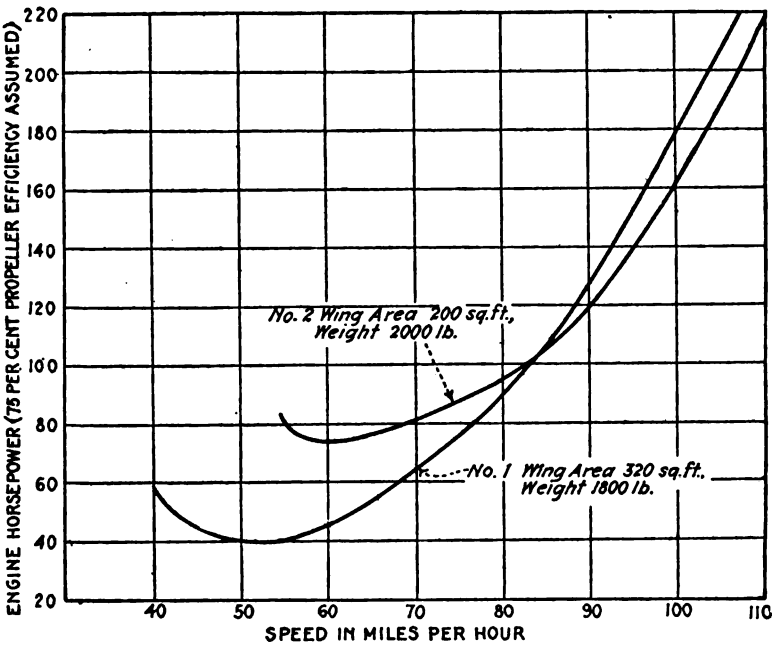

FIG. 1—POWER REQUIREMENTS OF TWO AEROPLANES

ation during the last year has shown clearly that no one size or type of engine can be considered satisfactory for all branches of military aviation. Non-military aviation is not yet sufficiently developed to standardize requirements, but they are undoubtedly not the same.

A matter that should not be overlooked by a designer is that engines in military service do not often have skilled attendance in the field. Delicate complicated parts are sure to give trouble. During the first months of the war many expert mechanics were recruited for the regular service, and when the military organization was completed, most of these men were either in the army or situated so that they

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were not available for aeronautic work. Power requirements frequently demand the use of twelve cylinders in spite of their many valves and spark-plugs needing care. Otherwise this type of engine is excellent, but it is not to be recommended for engines of less than 200 hp. The desirability of four valves per cylinder is questionable in any military engine; it is too much to expect field mechanics to keep all valves tight and in adjustment, in addition to all the other details that must be attended to.

NECESSITY FOR HIGH VELOCITIES

Most military aeroplanes are expected to make flights of several hours' duration over enemy land. Maximum speeds of less than 80 to 90 m.p.h. are almost sure to mean that an aeroplane will be brought down by enemy aviators in short order. Speeds of 90 to 100 m.p.h. are now becoming more frequent. The speeds will continue to increase as far as possible because of the great advantage possessed by a high-speed machine over its lower-speed adversary. The only reason these speeds are not exceeded is that the practical limit with the present type of construction is nearly reached.

The curve, Fig. 1, of horsepower requirements of two aeroplanes similar to the majority now in use, shows how rapidly the power required increases with speed. It is for this reason that few water-cooled engines of less than 140 hp. are used in aeroplane service. The power must be increased to 200 or 300 hp. and even more for the larger destroyer aeroplanes and seaplanes. Many seaplanes are in use with 300 to 350 hp., usually divided between two units. A large British triplane, in France, is propelled by 1400 hp. in four units. This triplane has a wing spread of 133 ft., and when carrying a load of five tons, has a total weight of 30,000 lb. The Curtiss Aeroplane Co. is understood to have had a number of triplanes of this type under construction for the British Government.

AEROPLANES FOR GUARDING CITIES

There is another type of aeroplane about which not much is heard because it is not built outside the countries engaged in war. Its sole duty is to protect large cities and other vulnerable points from enemy aircraft. It must be able to climb rapidly and possess high speed. Reliability is not so essential as for the raiders because it is always possible to land on home territory in case of engine trouble; fuel economy is not so important because it does not often remain in the air more than an hour. For such aeroplanes the revolving air-cooled engines of the Gnome and La Rhône types reign supreme because of their light weight. It is no secret that all altitude and speed records are held by engines of this type.

News has reached this country that the speed record of 126 m.p.h. established by Prévost in 1913 has been broken recently in France.

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There is little doubt as to the type of engine that could make this possible. So in spite of the rumors frequently reaching us that revolving engines are falling into disfavor, it must be realized that this is true only in certain classes of military work. They weigh but little more than half as much as the water-cooled types, and on short flights where the weight of fuel carried is not large, their inherently greater fuel consumption is not a handicap. For long flights, however, the increased weight of fuel that has to be carried is frequently more than the reduction of engine weight so that the weight of the total powerplant will be greater.

PROPELLER REQUIREMENTS

It does not seem logical for a designer to split hairs to get the greatest possible power from an engine and then allow it to be equipped with a propeller that cannot transmit more than 70 per cent of the power into useful work. The best efficiency realized by present propellers is about 75 per cent when they are properly selected, although

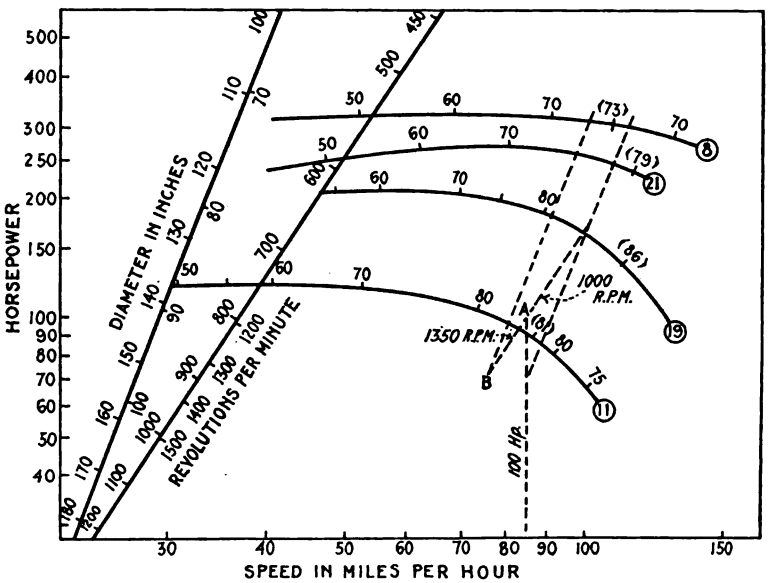

FIG. 2—CURVES SHOWING PROPELLER CHARACTERISTICS

in a few instances it is as high as 80 per cent. From recent experiments by Gustave Eiffel, (of Eiffel Tower fame), it seems possible that 85 per cent can be obtained in practice. The curves of propeller characteristics shown in Fig. 2 are taken from his experiments and are reproduced in order to give the designer a rational means of determining at what speed the propeller-shaft of his engine should run; also whether it is desirable to use a direct drive or a gear transmission with its added weight and 3 to 5 per cent loss of power.

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Each of the curves in Fig. 2 is for a different type of 100-in.-diameter propeller and shows the power absorbed at 1200 r.p.m. when advancing through the air at different velocities. The figures on the curves represent efficiencies. To use these curves for propellers of different diameters and at other speds than 1200 r.p.m., a point *A*, representing the coordinates of power and velocity, should be located (see Fig. 2) and another point *B* located on a line through *A*, parallel to the r.p.m. scale; the distance between the points being measured by the distance on the scale between 1200 r.p.m. and the required r.p.m. and in the same direction. Through *B* draw a line parallel to the diameter scale. The intersections with the curves then show the efficiencies to be expected under the imposed conditions. The diameters of the propellers for these conditions are found in a similar way on the diameter scale.

As an example suppose that it is desired to find the diameters and efficiencies of these different types of propellers turning at 1000 and 1350 r.p.m. for a 100-hp. engine in an aeroplane traveling 85 m.p.h. From the curves are obtained the results given in Table I.

TABLE I—PROPELLER DIAMETERS AND EFFICIENCIES

Propeller No.	Diameter, In.		Efficiency, Per Cent	
	1,000 r.p.m.	1,350 r.p.m.	1,000 r.p.m.	1,350 r.p.m.
11	115	92	80½	80½
19	101	83	84	81
21	93	77	78	76
8	89	74	72½	72

Notice that if 1350 r.p.m. represents the speed of a direct-driven propeller, and the use of a gear-driven propeller at 1000 r.p.m. is contemplated, there would be actually a loss in overall efficiency, when the gear loss is included, with propellers No. 11 and 8, and nothing would be gained with the others. Lower speeds require greater propeller diameters for the same power; and large diameters are not desirable since the propellers must be placed higher in order to clear the ground or water as the case may be. Gearing is always used, however, on engines running normally at speeds over 1800 r.p.m. When the propeller is above the center of resistance of an aeroplane, there is a destabilizing action when the engine is stopped or started.

FLYWHEEL EFFECT OF PROPELLER

The moment of inertia of a propeller is much higher than is usually supposed, and there is considerable flywheel effect as well as gyroscopic action. The gyroscopic action of a Gnome engine, about which many people worried in the past, is only about two-thirds as great as that of its propeller; gyroscopic action causes no annoyance to the pilot because he can easily counteract it by means of his controls. The correction is similar to that required to compensate for the torque reaction of an engine as it is stopped or started. The flywheel

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effect of a 9½-ft. propeller weighing 30 lb. is equivalent to that of a steel disk 3 in. thick and 22 in. diameter, weighing over 300 lb.

TYPES OF AEROPLANES AND ENGINES

The engine and radiator comprise the last unit of an aeroplane to conform to the stringent requirements of stream-line form, but since the high speeds necessitated in military work are forcing refinement, this point must be seriously considered in the future. Narrow width of engine and special radiators mounted at the sides of the body are a great advantage in this respect. With air resistance in mind, a comparison of the different engine installations shown in Figs. 3 to 23 will be instructive. Such a comparison will be more graphic if one realizes that a cylindrical body, with length equal to diameter,

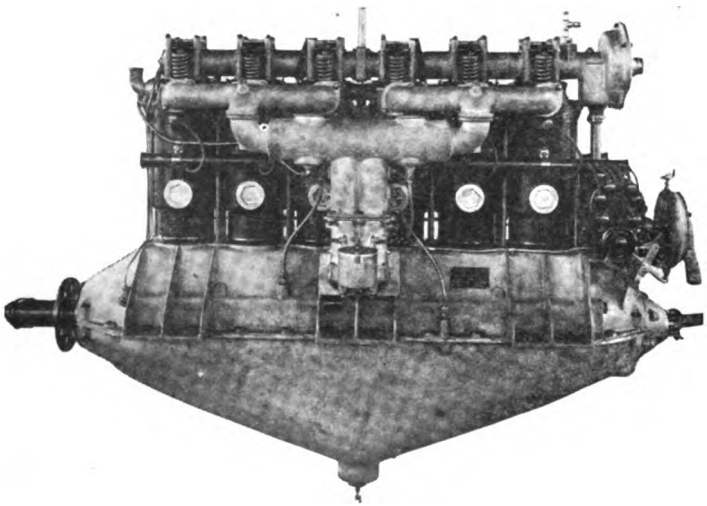

FIG. 3—HALL-SCOTT "A-5" ENGINE

Horsepower, 125; bore and stroke, 5 by 7 in.; r.p.m., 1250; weight, 525 lb.; weight per horsepower (including water and radiator), 6.2 lb.

offers about ten times as much air resistance as if enclosed in a fairly good stream-line casing of the same diameter and three times as long.

Hall-Scott "A-5" Engine

The cylinders of the Hall-Scott engine, Fig. 3, are of cast iron with integral water jackets; the pistons are also of cast iron. Notice the long studs that secure the cylinders. These studs are held on lugs cast near the top of the cylinders, the lugs being machined where they come in contact with those of adjacent cylinders, thus bracing the cylinders when the studs are tightened. The intake manifold is both water- and oil-jacketed. Double ignition is used. A detail of the valve gear of this engine is shown in Fig. 4.

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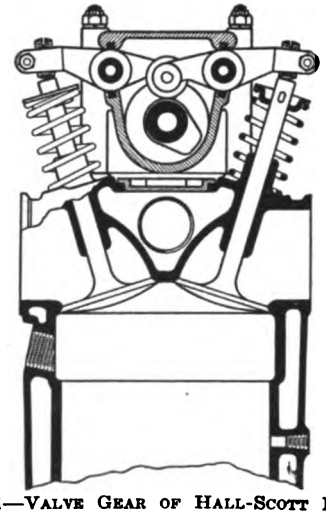

FIG. 4—VALVE GEAR OF HALL-SCOTT ENGINE

Sunbeam-Coatalen Engine

In the Sunbeam-Coatalen engine, Fig. 5, the compressed-air starting motor is placed just above the propeller hub.

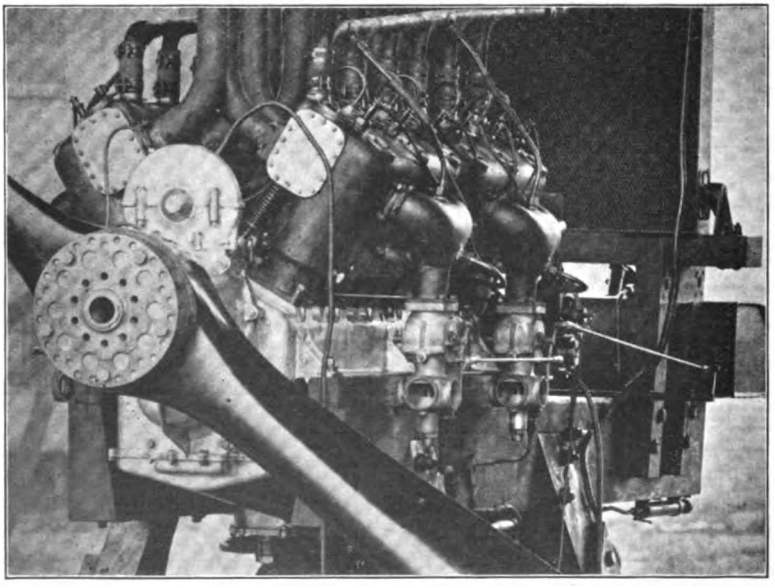

FIG. 5—SUNBEAM-COATALEN ENGINE

Horsepower, 225; bore and stroke, 3.54 by 5.91 in.; r.p.m. (engine) 2000, (propeller) 1000; weight, 905 lb.; weight per horsepower (including water and radiator), 4.8 lb.

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Curtiss "V-2" Engine and Military Tractor

The cylinders of the Curtiss "V-2" engine, Fig. 6, are of steel and the pistons of aluminum alloy. The "rat-trap" exhaust springs are interesting. Double ignition is used and two carbureters. This

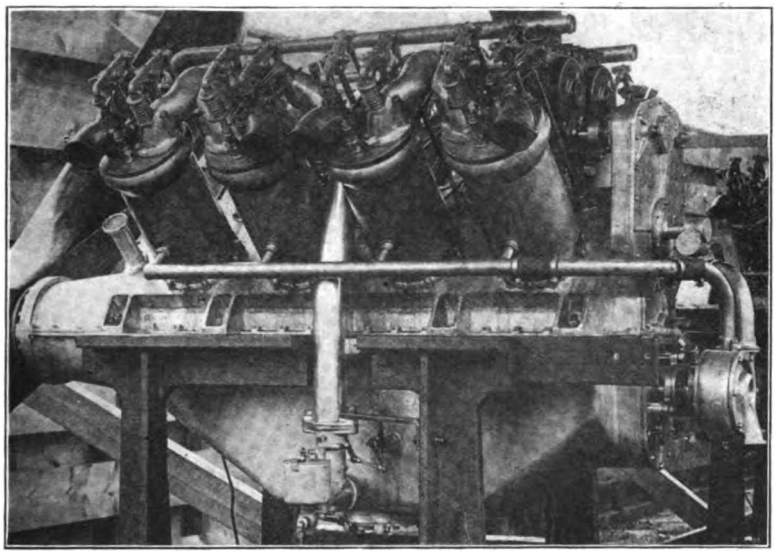

FIG. 6—CURTISS "V-2" ENGINE

Horsepower, 160; bore and stroke, 5 by 7 in.; r.p.m., 1300; weight, 565 lb.; weight per horsepower (including water and radiator), 4.4 lb.

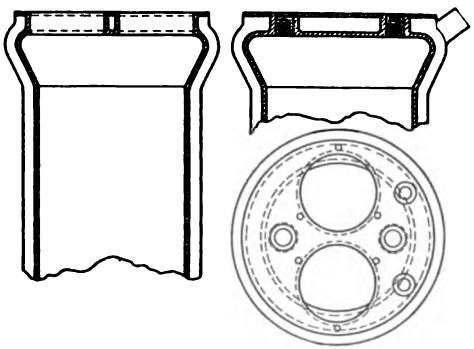

FIG. 7—STEEL CYLINDER OF V-2 CURTISS ENGINE

company is now developing a twelve-cylinder engine of about 225 hp. Fig. 7 shows the steel cylinders of the "V-2" engine.

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FIG. 8—CURTISS "J N" MILITARY TRACTOR

The location of the engines in the Curtiss military tractor, Fig. 8, is characteristic of Curtiss flying boats. For land machines, it has

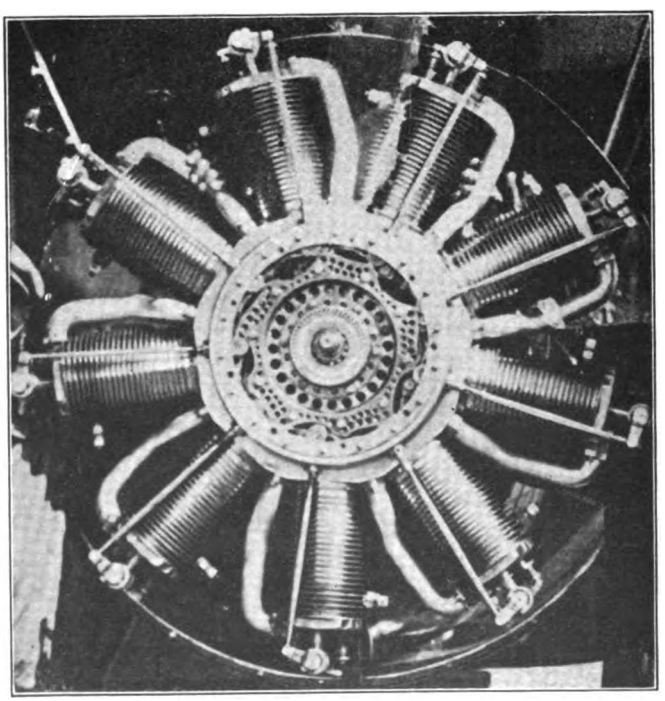

FIG. 9—LA RHÔNE REVOLVING ENGINE
Horsepower, 80; bore and stroke, 4.13 by 5.52 in.; r.p.m., 1200; weight, 253 lb.; weight per horsepower, 3.2 lb.

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the advantage that machine guns can be mounted to fire both fore and aft.

La Rhône Revolving Engine

The revolving air-cooled La Rhône engine, Fig. 9, now manufactured by the same company as the Gnôme, is generally considered to be superior to the latter. The cover of the crankcase and propeller hub is shown removed, thus exposing the cam mechanism. A single "push and pull" rod operates both valves of each cylinder. The valves are both in the cylinder head and are controlled by a rocker mounted on ball bearings so as to simplify lubrication.

Wisconsin Six-Cylinder Engine

In the Wisconsin engine, Fig. 10, the cylinders are of cast aluminum alloy, with steel liners. The valve seats are of cast iron, cast

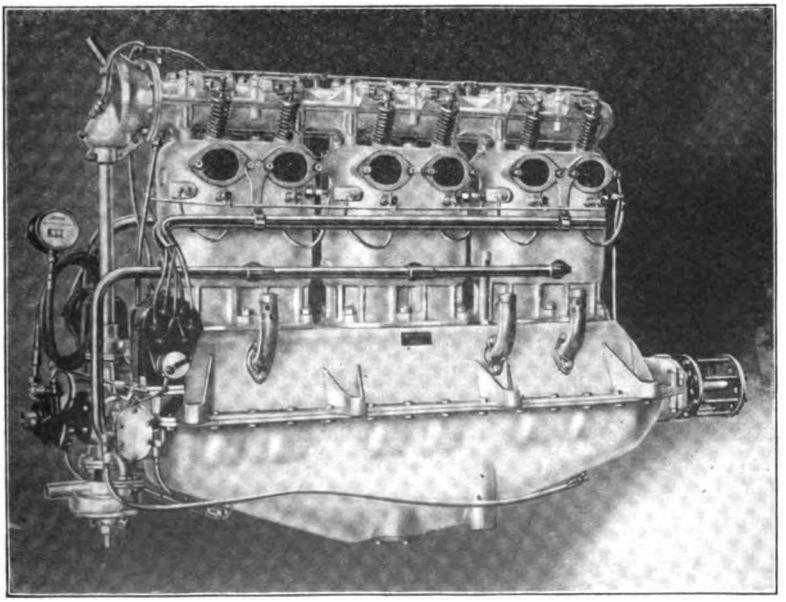

FIG. 10—WISCONSIN ENGINE (SIX-CYLINDER)

Horsepower, 140; bore and stroke, 5 by 6½ in.; r.p.m., 1400; weight, 600 lb.: weight per horsepower, 4.3 lb.

in place. The pistons are of aluminum also. Double ignition is used.

A twelve-cylinder Wisconsin engine similar to the six-cylinder model is also built. The horsepower of the twelve is 250; bore and stroke 5 by 6½ in.; r.p.m., 1200; weight, 1000 lb.; weight per horsepower, 4 lb. Opposite connecting-rods are pinned together at their large ends, as in the Renault engine.

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Mercedes Inverted Engine

The Mercedes engine, Fig. 11, is shown because of its originality. It was developed before the war in order that the aviator could have a better view. The construction lowers the center of gravity of the

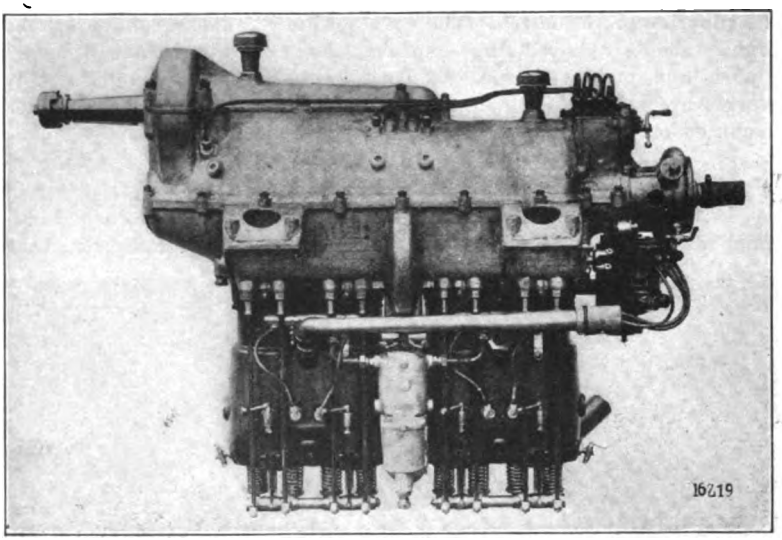

FIG. 11—MERCEDES INVERTED ENGINE
Horsepower, 72; r.p.m., 1412

aeroplane as well as raises its center of thrust. The cooling is ideal; cold water entering around the hottest part of the cylinders. Even the formation of steam would not cause trouble. Oil is supplied to the bearings under pressure, the leakage lubricating the cylinder walls.

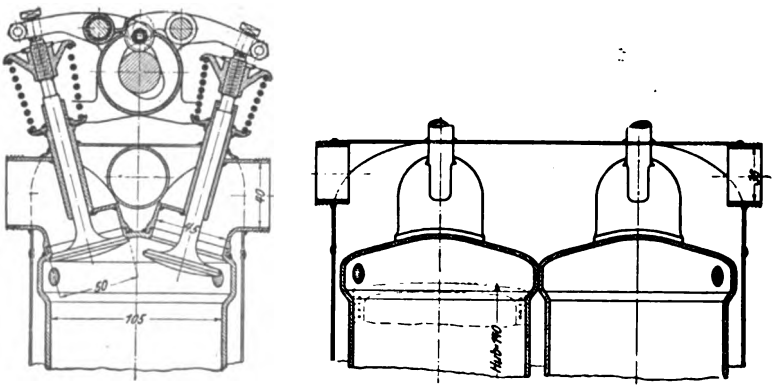

FIG. 12—MERCEDES VALVE GEAR AND STEEL CYLINDER

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The oil is used only once, fresh oil being supplied to bearings, and all that leaks past the pistons and is not burned passes out the exhaust. No failure of spark-plugs due to oil or carbon was experienced during the test for the Kaiser Prize, in which it won the fourth place. In Fig. 12 is given a detail of this engine showing how the jackets are built up by welding.

Sturtevant Engine and "Battleplane"

The Sturtevant eight-cylinder engine, Fig. 13, is the same as last year's model except for the gravity-feed carbureter and cast-aluminum intake-manifolds. A new model of the same type with aluminum cylinders and pistons will soon be announced. A twelve-cylinder engine is being developed also.

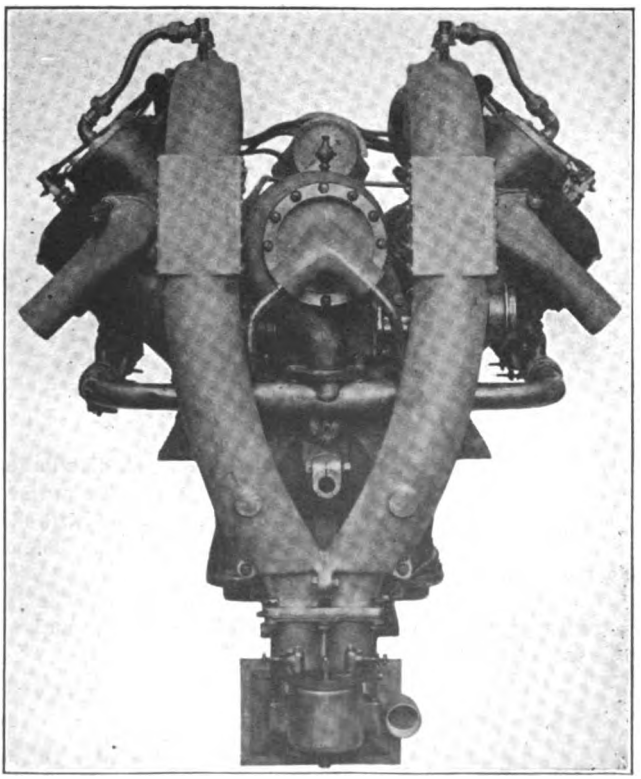

FIG. 13—STURTEVANT EIGHT-CYLINDER V-TYPE ENGINE

Horsepower, 140; bore and stroke, 4 by 5½ in.; r.p.m. (engine), 2000; (propeller), 1000 to 1500; weight, 580 lb.; weight per horsepower (including water and radiator), 4.9 lb.

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In the Sturtevant "battleplane," Fig. 14, the location of the gunners is unique. Most of this aeroplane is of steel; the body is of riveted steel angle-iron girder construction. The ribs and spars are also of steel.

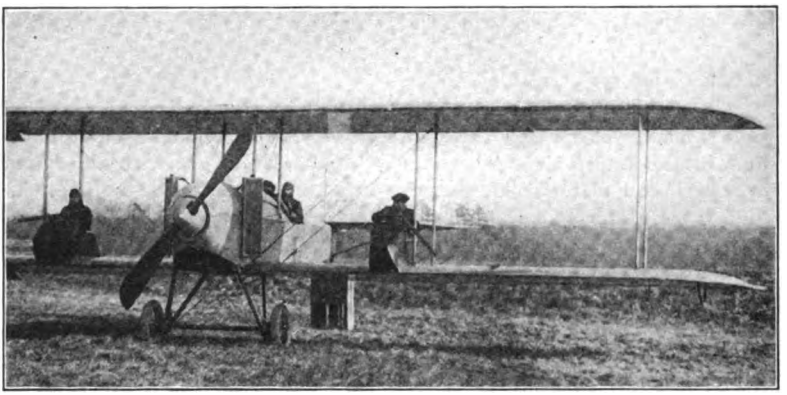

FIG. 14—VIEW OF STURTEVANT "BATTLEPLANE"

Thomas Engine and Military Tractor

In the Thomas eight-cylinder engine, Fig. 15, the distributor valve of the compressed-air starter is placed just above the water pump. The cylinders are cast in pairs, of cast-iron, and look a little like those in the Sturtevant engine. The pistons are of aluminum. The connecting-rods and cylinders are staggered.

The Thomas military tractor is shown in Fig. 16. The type and location of the radiator are typical of many different makes. This aeroplane makes 95 m.p.h. with a 135-hp. Thomas engine.

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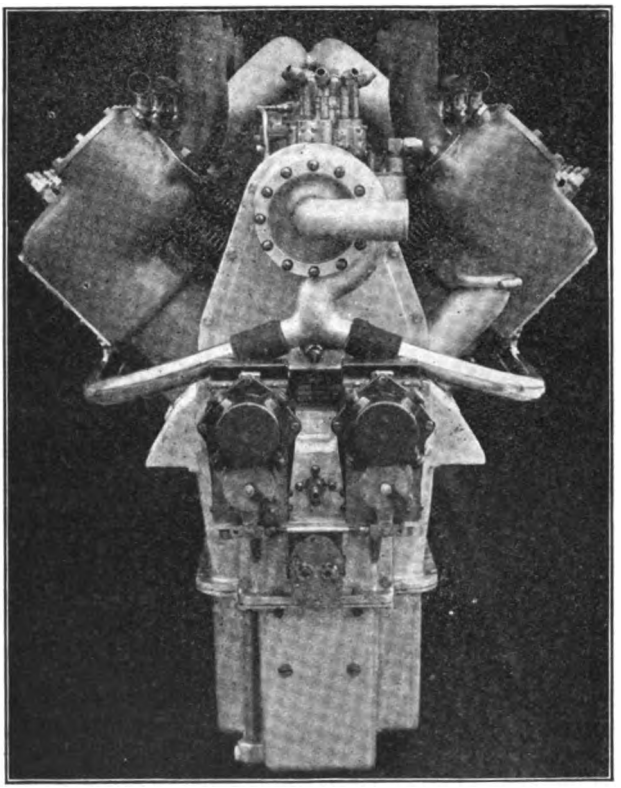

FIG. 15—THOMAS EIGHT-CYLINDER ENGINE

Horsepower, 135; bore and stroke, 4 by 5½ in.; r.p.m. (engine), 2000; (propeller), 1200; approximate weight per horsepower (including water and radiator), 4.9 lb.

FIG. 16—VIEW OF THOMAS MILITARY TRACTOR

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Martin Eight-Cylinder Engine

The Martin engine, Fig. 17, has four valves per cylinder, all four being operated by one pair of rockers, which are actuated by a single "push and pull" rod, as in the Austro-Daimler. A single camshaft is used for the valves of both rows of cylinders. The cylinders are in pairs, of steel welded together in somewhat the same fashion as the Mercedes steel cylinders. A pair of these cylinders, with studs, is

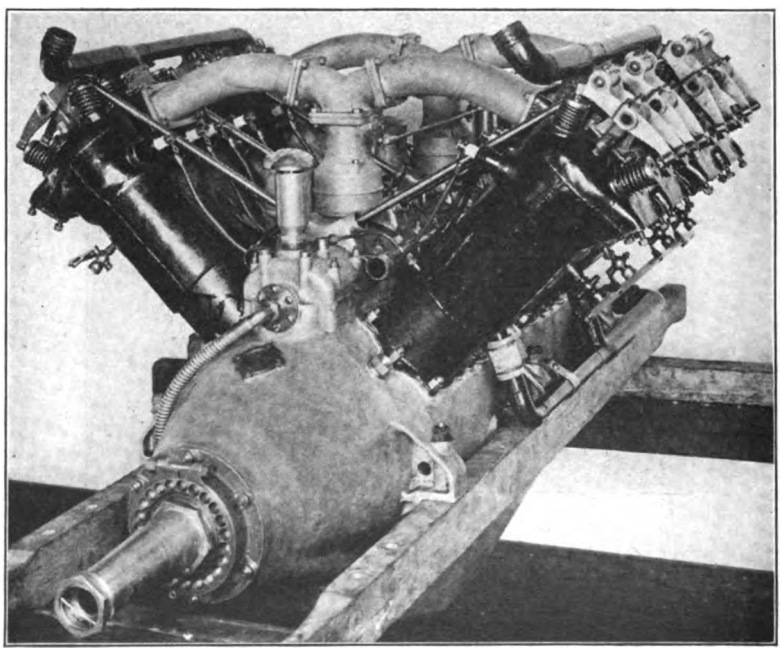

FIG. 17—MARTIN ENGINE OF EIGHT-CYLINDER V TYPE

Horsepower, 190; bore and stroke, 4⅝ by 7 in.; r.p.m., 1400; weight, 484 lb.; weight per horsepower (including water and radiator), 3.4 lb.

claimed to weigh but 24 lb. The pistons are aluminum alloy. The connecting-rods are of I-section and forked; hence the cylinders are not staggered. The lubrication is by pressure and splash, the two systems being independent. An interesting ignition system has been adopted, consisting of two single magnetos with a detached double distributor. Two-spark ignition is employed. Two carbureters and unusual manifolds are placed in the "V."

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Christofferson Six-Cylinder Engine

The intake manifold of the Christofferson engine, Fig. 18, is a noteworthy attempt to provide equal distribution from one carbureter to all cylinders. The cylinders are of steel with heads integral, the valves seating in the head. The jackets, in pairs, are of cast aluminum. The pistons are aluminum also. The valve mechanism is overhead and en-

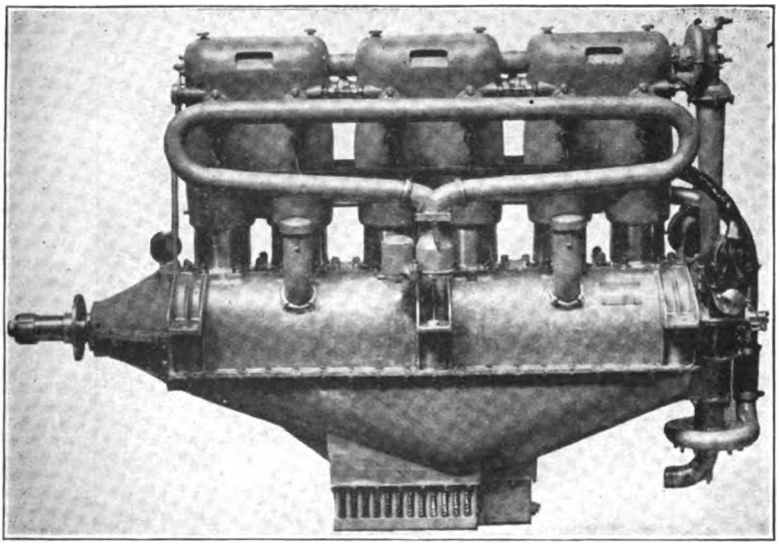

FIG. 18—CHRISTOFFERSON SIX-CYLINDER ENGINE

Horsepower, 120; bore and stroke, 4¾ by 6 in.; r.p.m., 1400

tirely enclosed. The camshaft is in three parts, connected between each pair of cylinders. The valve rockers, arranged as in the Hall-Scott and Mercedes engines, have fulcrums so located that the lift of cam is about one-half as great as the valve lift. Double ignition is used. Notice the oil radiator below the sump. A twelve-cylinder 240-hp. engine is being developed.

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Van Blerck Twelve-Cylinder Engine

The Van Blerck engine, Fig. 19, is constructed almost entirely of steel, including cylinders and crankcase. The water-jackets are spun copper; pistons are cast-iron; connecting-rods, of tubular section, bear on the same crankpin for opposite cylinders; hence cylinders are staggered. Double ignition and two twelve-cylinder magnetos are used.

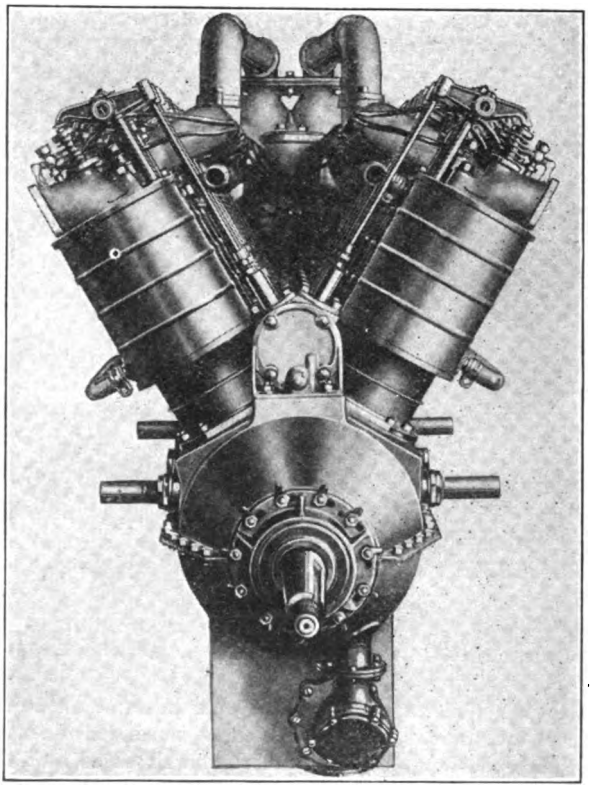

FIG. 19—VAN BLERCK TWELVE-CYLINDER ENGINE

Horsepower, 185; bore and stroke, 4½ by 5½ in.; r.p.m., 1400; weight, 600 lb.; weight per horsepower (including water and radiator), 4.1 lb.

Gallaudet "Destroyer Seaplane"

The Gallaudet "Destroyer Seaplane," Fig. 20, is an excellent design from the point of view of wind resistance. The four-bladed propeller and the radiators are placed in a unique location near the middle of the body. The two engines give a speed of 100 m.p.h.

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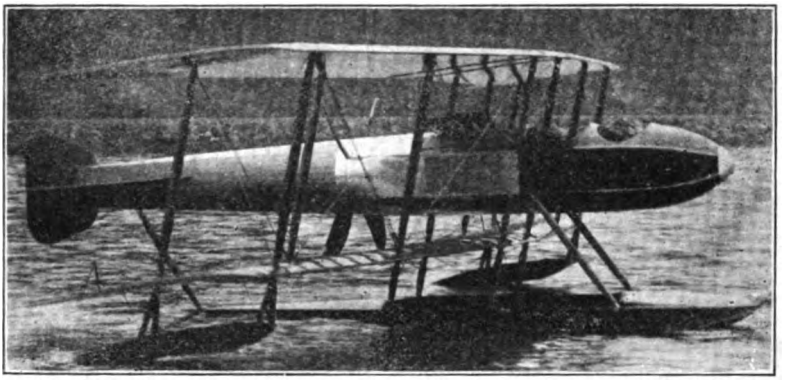

FIG. 20—GALLAUDET "DESTROYER SEAPLANE"

D. W. F. Biplane (German)

The D. W. F. biplane, Fig. 21, shows a characteristic mounting of the vertical engine in German aeroplanes, involving small head resistance. The radiator is on the other side of the body.

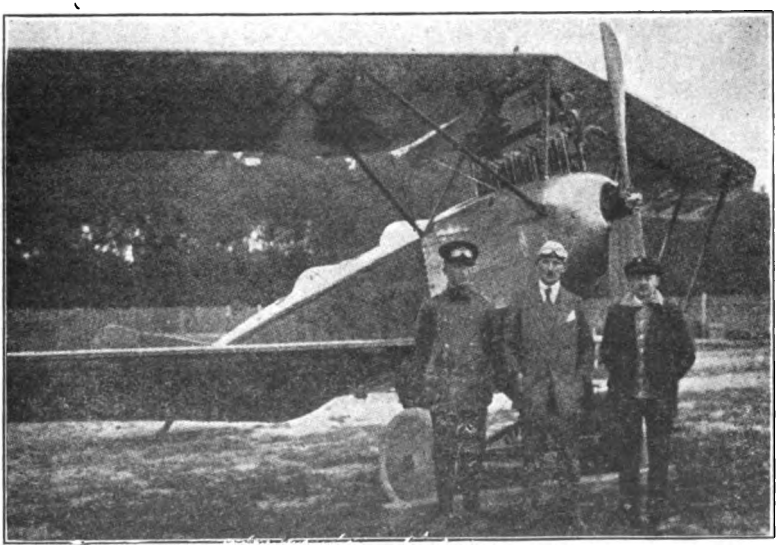

FIG. 21—D. W. F. BIPLANE WITH VERTICAL ENGINE

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"Fritz" Warplane With Two Engines

The "Fritz," Fig. 22, is a large double-engined German warplane that seriously embarrassed the Allies when it appeared on the front. A vertical Mercedes engine is mounted on each side of the body,

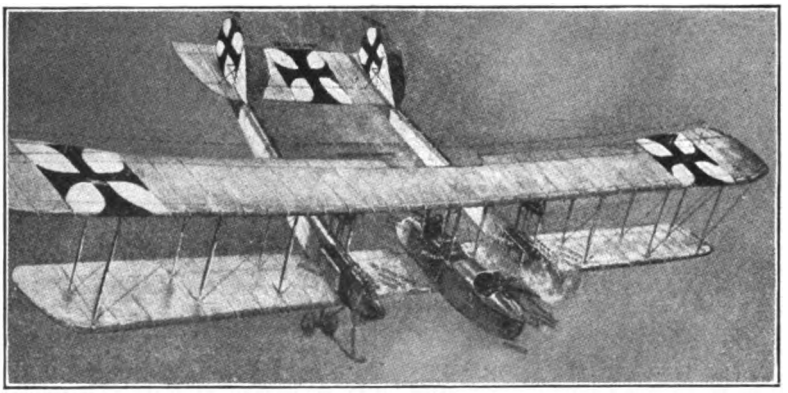

FIG. 22—"FRITZ" WARPLANE WITH TWO MERCEDES ENGINES

which is occupied by the gunners. The radiators can be seen at the sides of the bodies. Notice the small area of the powerplant offering head resistance.

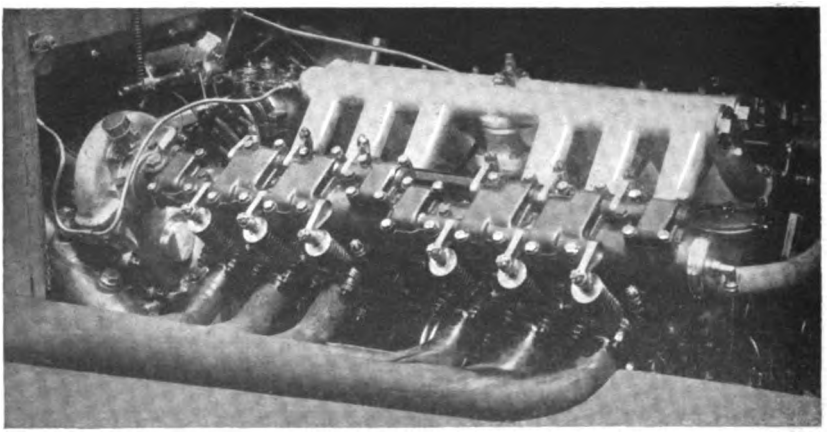

FIG. 23—PACKARD TWELVE-CYLINDER ENGINE

Horsepower, 100; bore and stroke, 2 21/32 by 4½ in.; r.p.m., 2300; weight, 500 lb.; weight per horsepower (including water and radiator), 5.9 lb.

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Packard Aeroplane Engine

In the Packard twelve-cylinder engine, Fig. 23, the construction of the valve-rocker housing to prevent oil leakage is ingenious; as are also the intake manifolds, which consist of a separate straight header for each row of cylinders with the carbureter at the center. Straight connections of equal length lead to the cylinders. The overhead camshafts are driven by trains of spur gears from the crankshaft. Aluminum pistons are used, and forked connecting-rods. The battery ignition and starting motor are of the "Delco" type. A four-valve twelve-cylinder 300-hp. engine is being developed with bore and stroke of 4 and 6 in. respectively. The general construction is almost identical with that of the 100-hp. engine.

DISCUSSION

GLENN MUFFLY:—The first engine of the revolving type was built on this side of the water by F. O. Farwell in 1898. The Adams-Farwell people made an engine of 5-in. bore on which the cooling ribs were entirely omitted. This was driven several thousand miles over bad roads, illustrating the fact that revolving engines can be air-cooled as satisfactorily as others are water-cooled.

The oil consumption of the old Adams-Farwell engine was low and it was an uncommon thing to find a spark-plug sooted, while the regrinding of valves was an operation actually unknown at the factory.

Since an engine must always run under full load in aviation work it should not have a speed of 2000 or 3000 r.p.m. Even higher speeds than these can be employed in an automobile engine, which is seldom run at its maximum speed, but the aviation engine is regularly operated under racing conditions.

Rotary valves will, I think, soon be used in aviation engines; in fact, I am now working on a revolving engine with that type of valve. This is to have a muffler, being one of the first aviation engines of the rotary type to be so equipped.

In speaking of the number of cylinders, Mr. MacCoull does not recommend the twelve for engines of less than 200 hp. It is necessary, however, to have a constant torque in an aeroplane engine in order to eliminate propeller flutter; thus it will be seen that there is an advantage in having more cylinders. A great deal of flywheel effect is due to the propeller, but intermittent torque causes a greater loss in aviation than in automobile operation.

FAULTS OF REVOLVING ENGINES

The main criticisms of the revolving type of engine—the two points on which it falls down—are excessive fuel and oil consumptions. Ten

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years ago we could get 17½ miles per gallon of gasoline with the old seven-passenger 50-hp. Adams-Farwell car, a performance which compares favorably with that of any of the cars of that time having reciprocating engines and even with those at present, for that weight of machine. The same car with five 5 by 5-in. cylinders would make over 400 miles per gallon of lubricating oil.

The consumption of lubricating oil in some revolving engines is excessive because of the way the oil is distributed. The steel cylinders and pistons require a great deal of clearance, thus allowing the oil to be thrown out through the exhaust ports by centrifugal force without accomplishing much lubricating or cooling. Centrifugal force can assist in the lubrication as well as in other functions of a revolving engine, but some designers seem to forget that it exists.

It is sometimes claimed that revolving engines are short-lived. This is not an inherent fault. At the Adams foundry a three-cylinder revolving automobile engine, built in 1903, ran ten hours a day for three years driving a cinder mill, washing iron out of the cupola drop. This engine required practically no attention, although it had previously served several years in an automobile.

The short life of most aviation engines is due to the extraordinary attempt to secure light weight and the failure to profit from automobile experience in design. Revolving engines have lost some of their prestige through over-anxiety of their designers to make them light. This type of engine is naturally lighter than the reciprocating engine; designers should not attempt to save a few additional pounds and thereby sacrifice fuel economy, which is now becoming important.

Most revolving engines swing in somewhat larger circles than necessary. We now have laid out on the drawing-board a 200-hp. engine that travels in only a 30-in. circle, thus reducing head resistance materially as compared with present aerial powerplants.

I prophesy that the revolving type of engine will in the near future have a fuel efficiency equal if not superior to that of the reciprocating type, and that it will be largely used in the automobile field as well as for aviation.

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COARSE CRYSTALLIZATION IN COLD-WORKED STEEL PARTS

By Ralph H. Sherry

(Detroit Section Associate)

Abstract

In this paper the author gives the results of an investigation of coarse crystallization. This investigation was carried out with commercial materials such as cold-drawn wire, hot and cold-rolled sheet, strip steel, cold-drawn tube and cold-pressings. The results of other investigations are briefly outlined.

Coarse crystallization, or grain-growth, it is stated, is due to the action of a limited amount of strain, exceeding the elastic limit, followed by annealing within certain temperature ranges. The experimental work which led to this conclusion is explained in detail in the paper. The effect of forging, cold-drawing, cold-rolling and cold-pressing was determined with commercial materials. Some study was made of the effect of carbon on grain-growth and of the effect of coarse crystallization on the physical properties.

In the discussion of commercial materials special reference is made to those used in motor car construction. Methods are given by which coarsely crystalline material can be detected. Methods of control which are necessary for the elimination of coarse crystallization are given in detail for each commercial material investigated, together with a general summary of the essential factors. The materials that receive individual consideration are sheet, strip steel, cold-drawn tube, cold-pressings and cold-drawn rivet stock.

The author's conclusions outline briefly the general conditions under which grain-growth occurs as determined in his investigation. Accompanying the article are tables giving in detail the results of cold-rolling and cold-drawing, together with other experimental data. A considerable number of microphotographs are given that illustrate the factors determined in the investigation.

RAW MATERIALS OF INDUSTRY

One of the most notable achievements of the automobile industry has been the development of highly specialized raw materials which can be fabricated into finished parts of great strength and lightness with a minimum loss in time, labor and waste. One thinks immediately in this connection of cold-pressed, cold-rolled or cold-drawn steel parts as paramount examples of this development. A great many classes of forgings and castings are being replaced by pressings, while automatic machine tools now make the production of many things from bar stock cheaper than from either forgings or pressings.

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It goes almost without saying that nearly all automatic machine stock must be cold-drawn if full production benefits are to be realized.

A great difficulty which has faced workers in these materials, and which has threatened to smother any success of the product by casting a suspicion as to quality over it, has been a marked tendency toward increased brittleness and coarse grain in cold-worked steel. From time to time mysterious epidemics of brittleness would crop out, bars and sheets breaking and cracking in fabrication, under ridiculously small stresses, with coarsely crystalline fractures. Annealing, strangely enough, often seemed only to aggravate the difficulty. Even pieces which came through manufacturing processes whole, were often put into assemblies under a shroud of fear that they would "crystallize in service."

Considerable attention has been given in recent years to the phenomenon of crystallization in pure iron and low-carbon steels subjected to strain, followed by annealing within certain temperature ranges. While crystallization, or grain-growth, can occur in certain other pure metals and homogeneous alloys under similar conditions, iron is worthy of special attention because of its extensive use and because of certain physical properties influencing grain growth, which it possesses.

Fortunately, most of the factors determining these irritating conditions have gradually become known and it is the purpose of this paper first to review the investigations covering this subject. The conditions under which crystallization or grain-growth takes place will then be explained. Its occurrence in specific products of interest to automobile engineers will be discussed. Methods by which this trouble can be prevented or cured will be outlined for each material investigated, and general conclusions with regard to the phenomena noted in the investigation will be given.

INVESTIGATIONS OF CRYSTALLIZATION

The occurrence of coarse crystallization in low-carbon steel at low temperatures was first mentioned by Stead (1)*, who observed it "in certain steels and pure irons, originally of fine grain produced by forging or by certain heat treatment," when annealed between 500 deg. C (930 deg. F.), where slow growth occurred, and at 600 to 750 deg. C (1110 to 1380 deg. F.), where the growth was more rapid. Annealing at 700 deg. C. (1290 deg. F.) for a few hours produced exceedingly coarse grains. Grain-growth was also observed in sheets containing 0.05 to 0.12 per cent carbon. One peculiarity noted was that grain-growth was never observed in sheets less than 18 gage in thickness (2). Experiments to produce the structure at will were not invariably successful.

*Numbers in parentheses refer to bibliography at conclusion of paper.

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Charpy (3) showed that a coarse granular structure could be produced in steel low in carbon and phosphorus by pressing the material in a screw plate and subsequently annealing at 700 deg. C (1290 deg. F.). He obtained grains about ten times the original size; by proper selection of the annealing temperature and time, extremely large grains could be produced.

Sauveur (4) produced the same condition in low-carbon steel, straining the metal by tension, compression, bending, and in the Brinell hardness testing machine subsequently annealing his specimens below the thermal critical range. Sauveur came to the conclusion that a definite or "critical" strain was necessary in order to produce grain-growth on annealing. In other investigations the strain necessary to produce grain-growth has been found to cover a rather wide range. The narrow limits noted by Sauveur are probably due to his use of annealing temperatures below the thermal critical range.

Chappell (5) investigated the phenomenon in somewhat the same way as Sauveur, applying strain along bars 12 centimeters in length, heating one end and keeping the other cool in the air, precautions being taken to ensure a regular fall of temperature along the bar. As a test of the quantitative effect of strain or deformation, tension was applied to tapered bars which were subsequently annealed at various temperatures. The effect of strain at high temperatures was investigated and exceedingly coarse grains produced by applying strain at 870 deg. C. (1600 deg. F.) and annealing later at about 850 deg. C. (1560 deg. F.). No occurrence of coarse crystallization was noted in annealed cold-drawn wire, this condition being produced only in pure iron that had received local deformation. It is stated, however, that the negative result is probably due to the wire having been strained beyond the limiting range. The effect of bending under shock was also tested. In cases of local deformation Chappell notes the presence of three zones after annealing within a certain range: one representing a high degree of deformation in which there is no appreciable change in the crystals; a central zone of moderate deformation in which there is a large crystalline growth; and a third zone of little or no deformation in which there is no growth, corresponding closely to the result obtained by Sauveur.

Some years ago the author had occasion to investigate the occurrence of coarse crystallization in low-carbon steel in various commercial forms and published a brief account (6) of part of this work. This investigation has been continued with several commercial varieties of cold-worked low-carbon steel, largely under commercial conditions.

The materials investigated were hot-rolled rod, cold-drawn wire, hot- and cold-rolled sheet, cold-rolled strip, cold-drawn tubing and pressings—the principal commercial forms of low-carbon steel in which coarse crystallization occurs. The effects of forging hot and cold were investigated on a small scale by the use of a hand hammer

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on small test-pieces. The quantitative effect of strain received special attention.

CONDITIONS UNDER WHICH GRAIN-GROWTH OCCURS

The formation of coarse grains in low-carbon steel will follow the action of a limited amount of strain exceeding the elastic limit and subsequent annealing within certain temperature ranges. This has been confirmed by all investigators of this phenomenon. In the author's earlier investigations certain other phenomena depending upon the quantitative effect of strain were observed and have been confirmed by the later investigations. On varying by narrow steps the amount of strain within the range mentioned, it was noted that the less the applied strain the greater was the grain-size on subsequent annealing. Within the limiting range of strain and near the lower limit there seems to be a definite or "critical" strain below which the range of temperature within which grain-growth will occur is limited by the Ar_1, Ar_2 thermal critical points—about 690 and 780 deg. C. (1275 and 1435 deg. F.). When the material is strained beyond this point, the annealing range is considerably extended, falling between about 650 and 900 deg. C. (1200 and 1650 deg. F.) The moderate grain-growth occurring below 650 deg. C. (1200 deg. F.) noted by other investigators was not found in this investigation. This "critical" strain has not been noted by other investigators, but the quantitative effect of strain on the grain-size has been partly confirmed by the work of Chappell.

STANDARDS OF MEASUREMENT

In order to study the effect of strain quantitatively it was necessary to find some fairly accurate and convenient standard of measurement for permanent deformation, direct if possible, but at least relative. Measurement of the strain applied beyond the elastic limit is of course direct, but is applicable only when the strain is applied in tension or compression in such a way that this may be determined, and, in commercial cases particularly, offers neither a convenient nor a practical standard. On this account the more convenient and relatively accurate reduction of area was adopted. This can be determined easily and with reasonable accuracy in a large number of commercial operations, as for instance the rolling of strip steel or the drawing of wire, where the measurement of the applied tensile strain would be impossible. As it is necessary to exceed the elastic limit in order to produce grain-growth, the reduction of area, with its origin at the elastic limit, is also a more relatively accurate standard of measurement. In certain cases, of which some pressing operations to be discussed are typical, the nature of the applied strain may be so complicated that no simple standard of measurement will serve to define it, and some purely relative and empirical standard must

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be adopted. In many such cases, however, the reduction of area will serve as a basic standard.

EXPERIMENTAL OPERATIONS

Forging and Hot-Rolling

A number of samples of hot-rolled rod of about ½ in. diameter, found to be fine-grained after annealing within the critical range, were forged under widely varying conditions, annealed and examined. The forging was carried on by means of a hand hammer and all the samples were finally annealed at 700 deg. C. (1290 deg. F.). Certain samples were heated to temperatures above 900 deg. C. (1650 deg. F.) and hammered, the temperature at which work ceased being estimated by some approximate means such as color in semi-darkness or by the scorching effect on wood. Other samples were hammered cold. Where grain-growth was found it was usually confined to a narrow area about 1/16-in. wide at the periphery of the piece. As nearly as could be determined by the rather crude method of measurement, no grain-growth occurred unless the rod was forged below a temperature somewhere between 650 and 750 deg. C. (1200 and 1380 deg. F.). Some of the results are given in Table I, which shows the grain size at the edge after one hour at 700 deg. C. (1292 deg. F.).

TABLE I—FORGING TESTS ON HOT-ROLLED ROD

	Grain Size, Mm.
Rod diameter 7/16 In.	
Original rod	0.02
Forged cold	0.40 to 0.16
Forged cold	0.30 to 0.18
Forged from 900 deg. C. (1650 deg. F.) to red	0.02
Same. Spot next to anvil	0.22 to 0.40
Forged from 900 deg. C. (1650 deg. F.) to black	0.14 to 0.22
Forged from 900 deg. C. (1650 deg. F.) to about 500 deg. C. (930 deg. F.)	0.14 to 0.40
Rod diameter 9/16 In.	
Original rod	0.02
Forged cold	0.22 to 0.40
Forged cold	0.18
Forged from 900 deg. C. (1650 deg. F.) to red	0.02
Forged from 900 deg. C. (1650 deg. F.) to black	0.18 to 0.40
Forged from 900 deg. C. (1650 deg. F.) until cold	0.10
Forged from 900 deg. C. (1650 deg. F.) until cold	0.08

No case of grain-growth was observed in annealed samples of hot-rolled rod, of which a considerable number were examined.

Cold-drawing

Samples of hot-rolled rod of from ¼ to ½ in. diameter were cold-drawn to various sizes and samples from each draft annealed at temperatures between 650 and 900 deg. C. (1200 and 1650 deg. F.).

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TABLE II—GRAIN SIZES OF ANNEALED COLD-DRAWN RODS

Dia., In.	Dia. After Draw, In.	Red. Area, per Cent	Grain-Sizes, Mm. 1 Hr. at 700 Deg. C.	Grain-Size, Mm., 1 Hr. at 800 and 850 Deg. C
0.386	0.372	7	0.02	0.02
0.386	0.368	9	Half 0.02, half 0.30 to 0.52	0.02
0.386	0.3645	11	0.27, a few 0.02	0.27 to 0.02
0.386	0.3595	13	0.15	0.15
0.312	0.302	7	0.52	0.02
0.312	0.296	10	0.10	0.10
0.312	0.289	15	0.06	0.06
0.312	0.280	20	0.04	0.04
0.312	0.271	25	0.036	0.036
0.384	0.336	7	0.02	0.02
0.384	0.334	8	0.50	0.02
0.384	0.332	9	0.40	0.02
0.384	0.330	10	0.22	0.22
0.351	0.342	5	0.02, a few 0.12	0.02, a few 0.12
0.351	0.339	6	0.02	0.02
0.351	0.338	7	0.52 to 0.72, a few 0.02	0.02
0.351	0.336	8	0.28 to 0.50	0.02
0.351	0 334	9	0 40	0.20 to 0.40
0.385	0.375	5	0.02	0.02
0.385	0 373	6	0.02	0.02
0.385	0.371	7	0.02, a few 0.20	0.02
0.445	0 429	7	0.02, some 0.14	0.14 to 0.02
0.372	0.361	6	0.02	0.02
0.351	0.334	9	0.50	0.02
0.351	0.324	15	0.14	0.14
0.351	0.315	20	0.05	0.05
0 351	0.304	25	0.03	0.03
0.437	0 415	10	0.20	0.20
0.437	0.404	15	0.12	0.12
0.437	0.392	20	0.04	0.04
0.437	0.379	25	0.03	0.03
0.486	0.452	7	0 02	0.02
0.486	0.445	10	0.27	0.27 to 0.02
0.486	0.432	15	0.12	0.12
0.486	0.419	20	0.05	0.05
0.439	0.416	10	0.40	0.02
0.439	0.410	15	0.17	0.17
0.439	0.392	20	0.05	0.05
0.498	0.472	10	0.22 to 0.35	0.22 to 0.35
0.498	0.460	15	0.11	0.11
0.498	0.450	20	0.05	0 05

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No grain-growth was found outside of this temperature range. In a number of tests the amount of reduction was varied by as narrow steps as possible, in some cases 0.001 in. For reasons which will be made evident, the samples in each set which were annealed at 700 deg. C. (1290 deg. F.) were selected for the determination of the quantitative effect of strain. Some of the results are given in Table II. From these it will be seen that, within certain limits, the grain-size after annealing is inversely proportionate to the applied strain.

On examining the samples annealed at various temperatures between the limits mentioned, it was noted that while the more heavily strained specimens showed a grain-growth after annealing within

TABLE III—GRAIN SIZES IN ROD OF 0.320 IN. DIAMETER

After Drawing		Grain Size in Mm. After 1 Hr. at	
Diameter, Inches	Red. of Area, per Cent	700 Deg. C. (1290 Deg. F.)	800-850 Deg. C. (1470-1560 Deg. F.)
0.309	6.5	0.02	0.02
0.307	8.	Some 0.02; rest 0.40 to 0.80	0.02
0.305	9.	0.28 to 0.56; a few 0.02	0.02
0.303	10.	0.25 to 0.40; a few 0.02	0.02; some 0.18
0.302	11.	0.20 to 0.25	0.20 to 0.25
0.300	12.5	0.15	0.15
0.297	16.	0.07	0.07

this limiting temperature range, some of these which were more lightly strained became coarsely crystalline only when annealed between about 690 and 780 deg. C. (1275 and 1435 deg. F.). Reheating above the latter point would refine any coarse-grained structure that might have been produced in the specimens. Photomicrographs of one of these specimens annealed at various temperatures between 675 and 780 deg. C. (1250 and 1435 deg. F.) are shown in Plate I. A large number of specimens of this material were examined after annealing at the higher temperature and all were found to be refined with the exception of one small section found on repolishing in which a few moderately coarse grains existed. This section was selected for photographing in order to show the narrow range of strain within which perfect refining at this temperature probably occurs.

The analysis of this steel is carbon, 0.08; manganese, 0.34; sul-

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phur, 0.032; phosphorus, 0.010, and silicon, 0.020 per cent. The reduction of area was probably about 9 per cent.

These results seem to indicate the existence of a "critical" strain affecting the annealing temperature ranges within which grain-growth occurs.

For further confirmation a number of other samples were drawn with varying reductions of area in narrow steps and annealed at various temperatures between about 650 and 900 deg. C. (1200 and 1650 deg. F.). The results obtained with one series are given in Table III and others in Table II.

The microphotographs of this series shown in Plates II to VI illustrate the decrease in the grain-size with increase in the applied strain as found in the series annealed at 700 deg. C. (1290 deg. F.). Smaller sections of the samples drawn with 8, 9, 10 and 11 per cent reduction of area and annealed at 780 deg. C. (1435 deg. F.) are also shown to illustrate the action of the "critical" strain. It will be noted that the samples show complete refining at this temperature until the reduction of area rises to about 10 per cent, when some coarse crystallization still remains. With heavier reductions no refining occurs until the material is annealed at about 900 deg. C. (1650 deg. F.).

Some difficulty was encountered in duplicating the results, as the rod was frequently out of round slightly and sometimes varied locally in diameter. The greatest variations were noted in attempting to determine the lowest reduction which would cause grain-growth. Where light reductions were used the effect was frequently confined to the periphery or to local areas, the remainder of the section being unaffected. An example of this will be seen in Plate II, showing a specimen annealed at 700 deg. C. (1290 deg. F.).

Occasionally sections were found in which the strain was unevenly distributed, as shown by the wide variation in the grain-size produced. In some lightly-strained samples which apparently became refined completely at 780 deg. C. (1435 deg. F.), a certain "skin" effect was sometimes noted. On examining longitudinal sections of samples annealed at this temperature, a thin layer of moderately coarse grains, possibly 0.12 to 0.18 mm. in diameter, was sometimes found close to the periphery, the rest of the piece being completely refined. Allowing for these minor variations and using proper precautions, it was generally possible to reproduce these results at will. With the exception of the local effects noted, no grain-growth occurred if the reduction of area fell below about 7 per cent.

In the tabulated results some variations between the different series will be noted. These are generally due to the previously-mentioned variation in the conditions of drawing.

Cold-Rolled Strip

A few samples of hot-rolled strip about 0.065 in. thick were cold-rolled with varying reductions of area. The same variation in grain-

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size with reduction was found with this material as with the cold-drawn steel. Refining took place at 900 deg. C. (1650 deg. F.) only, no "critical" strain being noted. The material was too light for careful measurement and the presence of certain irregularities in the surface made impossible an exact determination of the reduction of area. The largest grain-growth produced was much less than that found in the cold-drawn specimens which had received comparatively light reduction. Some of the results obtained are given in Table IV.

TABLE IV—COLD-ROLLED STRIP SHOWING QUANTITATIVE EFFECT OF STRAIN

Size After Cold Rolling, In.	Red. of Area, per Cent	Grain Size in Mm. After One Hour at 700 to 870 Deg. C. (1290 to 1600 Deg. F.)
	Hot-rolled Strip 0.0625 *in. thick*	
0.059	5.6	0.034, a few 0.10
0.0575	8.	0.03, few 0.14 to 0.27
0.056	10.	0.14 to 0.28
0.0545	13.	0.09 to 0.14
0.053	15.	0.07 to 0.11
	Hot-rolled Strip 0.064 *in. thick*	
0.061	4.5	0.02 to 0.03
0.0585	8.5	0.02 to 0.27
0.0575	10.	0.21, some 0.02
0.056	12.5	0.14
0.054	15.5	0.10
0.0515	19.5	0.05 to 0.08
0.049	24.5	0.027 to 0.036

No heavier material was available at the time the investigation was made. Further study was confined to a number of cases of coarse crystallization found in strip 1/10 to ⅛ in. thick, obtained from various sources.

A few cases were found in which complete refining occurred on annealing at 780 deg. C. (1435 deg. F.). In other cases, partial refining occurred at this temperature. In a few cases no refinement occurred until the material was annealed at 900 deg. C. (1650 deg. F.). In the cases where complete or partial refinement occurred at 780 deg. C. (1435 deg. F.) the original grain-size varied from 0.52 to 0.65 mm. in diameter. Where 900 deg. C. (1650 deg. F.) was required the grain-size was smaller than this.

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Cold Pressings

The quantitative study of grain-growth in low-carbon steel pressings is decidedly difficult if the nature of the applied strain is at all complicated. In certain cases the amount of strain can be determined relatively as, for instance, in operations such as bending, light drawing and forming. In a drawing operation the amount of strain can be determined relatively if carried out with no reduction in the thickness of the metal.

In Fig. 1 the outer circle represents the circumference of the blank, the inner circle the circumference of the pressing. In the pressing operation the periphery of the blank becomes the top of the pressing

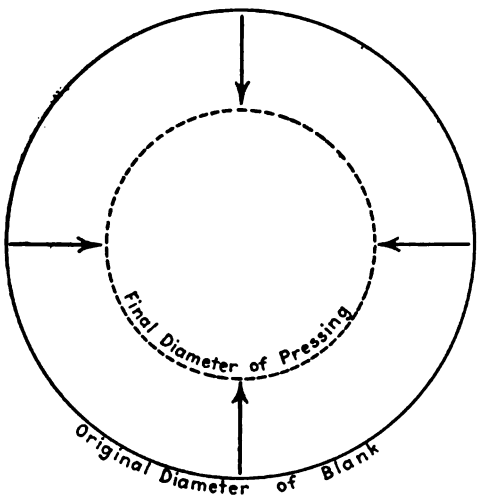

FIG. 1—OUTLINES OF BLANK AND OF PRESSING

and receives the heaviest strain. The bottom of the pressing receives little or no strain. In the side walls are zones in which the strain increases regularly from the bottom to the top. In the drawing, Fig. 1, these zones are included in the area between the inner and outer circles. The strain applied over these zones can be represented relatively by the frustum of a cone of the same height as the pressing, the radii of the upper and lower bases being respectively equal to the radius of the blank and the radius of the pressing. In such a figure the radius of the smaller base represents the zero point of measurement. The difference of the radius of the smaller base and the radius of the larger base represents the maximum amount of strain applied to the pressing.

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Fig. 2 is a diagrammatic representation of the relative strains applied in a typical pressing operation. If such a pressing is annealed at 700 deg. C. (1290 deg. F.) and examined, it will be found that there has been considerable grain-growth, large at the bottom and decreasing in size regularly toward the top.

Strips were cut from a number of such pressings and annealed at various temperatures between 600 and 900 deg. C. (1110 to 1650 deg. F.). On examining samples annealed between 690 and 780 deg. C. (1275 and 1435 deg. F.) it was noted that a sharply defined zone of marked grain-growth occurred near the bottom of the pressing. Near the top of the pressing a zone was found in which the grain-size was normal. In the central zone extending between the bottom and top

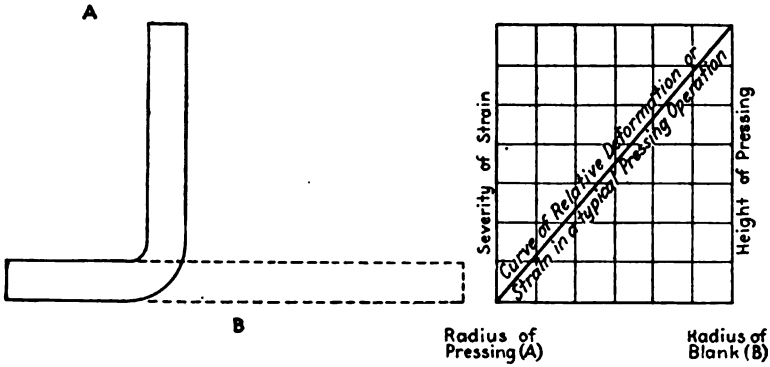

FIG. 2—STRAIN IN TYPICAL PRESSING OPERATION

zones the grain-size decreased gradually until it became normal. No grain-growth occurred at the bottom. On reheating to 790 deg. C. (1450 deg. F.) partial refinement took place in the lower zone. Samples annealed between 780 and 875 deg. C. (1435 and 1605 deg. F.) showed rather moderate grain-growth in the lower zone, decidedly less than that obtained on annealing between 690 and 780 deg. C. (1275 and 1435 deg. F.). On annealing below 650 deg. C. (1200 deg. F.) for periods of time up to six hours, no grain-growth was found. On annealing between about 650 and 690 deg. C. (1200 and 1275 deg. F.) no grain-growth occurred in the lower zone and only a moderate growth in the upper zone. The microphotographs in Plate VII illustrate the effect of these varied annealing temperatures. In preparation for these, parallel strips were cut through the pressing and annealed at the temperatures noted. The microphotographs represent the same relative positions in the pressing. No variation for increase in time was noted.

A similar condition was noted on experimenting with a number of other pressings of widely varying gage, which had received rather

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moderate drawing, bending or forming. These results tend to confirm the existence of the "critical" strain noted in the experiments with cold-drawn material. That refining occurs at about 780 deg. C. (1435 deg. F.) was noted in actual practice with many pressings in which the presence of coarse grains prevented any further operations. After annealing at about 800 deg. C. (1470 deg. F.) the metal could be readily worked. In a comparatively small number of cases where the operation expected was somewhat severe, it was necessary to anneal at 900 deg. C. (1650 deg. F.).

EFFECT OF CARBON

The observations of nearly all the investigators of the phenomenon under discussion tend to show that grain-growth will not occur in steel that has a carbon content uniformly above 0.15 to 0.18 per cent.

TABLE V—PROPERTIES OF COARSE-GRAINED HOT-ROLLED SHEET

Thickness, In.	E. L., Lb. per Sq. In.	T. S., Lb. per Sq. In.	Elong., 2 In., per Cent	Grain Size, Mm.
0.096	23,000	46,000	33	0.20
0.093	26,300	50,000	36	0.10
0.095	23,000	47,200	38	0.15
0.074	24,000	45,000	36	0.08
0.064	24,600	46,000	36	0.10
0.067	26,000	48,700	35	0.12
0.122	24,700	47,000	36	0.12

While no exact determination was made, this was practically confirmed in this investigation, although a number of cases of grain-growth in steel of from 0.18 to 0.20 per cent carbon were observed where ferrite bands were present. A microphotograph of a case of this kind in a hot-rolled sheet, which contained 0.18 per cent carbon, is shown in Plate VIII. No case of grain-growth was observed in steel uniformly 0.18 per cent or more in carbon content.

PHYSICAL PROPERTIES

While the most noticeable difference in the physical properties of normal and coarse-grained low-carbon steel is in resistance to shock, some marked differences were noted at times in the tensile properties. While these may vary in sheet according to the conditions of rolling, the presence of coarse grains seemed generally to lower the elastic limit and the elongation. Some instances are given in Table V. Under normal conditions the physical properties of the hot-rolled

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sheet were: Elastic limit, 30,000 to 35,000 lb. per sq. in.; tensile strength, 45,000 to 50,000 lb. per sq. in.; elongation in 2 in., 42 to 52 per cent.

Duplicate samples from the same sheet usually gave similar results, but after annealing at 900 deg. C. (1650 deg. F.), the tensile properties became quite normal. Similar properties were noted in coarse-grained cold-drawn and cold-rolled material.

EFFECT OF TIME OF ANNEALING

The effect of increasing the length of time of annealing was tried with all the material investigated, but no increase in the grain-size was found because of prolonged annealing. Samples were annealed for varying periods of time up to eight hours. Some difference in the time required was noted with cold-drawn steel and the light pressings. One-half hour was required to produce grain-growth in the drawn material, while in thin pressings a few minutes was sufficient.

THERMAL CRITICAL POINTS

Thermal critical points were determined with a coarse-grained sample from a large pressing, the stock being ½ in. plate, basic open-hearth, 0.05 per cent carbon. The Leeds and Northrup transformation point apparatus was used for this purpose, the resistance being kept low in order to make small changes noticeable. A slight change was noticed at the A_1 points. At about 775 deg. C. (1430 deg. F.) a marked change was found on the first heating curve, the differential marker leaving the field entirely. The Ac_3 point was very marked at about 910 deg. C. (1670 deg. F.). On cooling the transformation points were not so marked. The Ar_3 point appeared at about 880 deg. C. (1615 deg. F.) and the Ar_2 at about 775 deg. C. (1430 deg. F.). The sample was reheated several times to note changes that might occur in the location or the intensity of the transformation points. The intensity of the points Ac_2 and Ac_3 was much diminished on the second heating, complete refining having taken place on the first trial. The intensity of the Ar points on cooling remained about the same. The location of the various points was unchanged.

COARSE CRYSTALLIZATION IN COMMERCIAL MATERIALS

Sheet and Strip Steel

The occurrence of a coarse-grained structure is common in sheet, both in hot- and cold-rolled annealed. In hot-rolled sheet no grain-growth was found in sizes heavier than 0.130 in., but it seemed to be rather common in all thinner sheet, including 0.065 in., the smallest size examined. This grain-growth was in most cases moderate, the grain-size usually being between 0.08 and 0.12 mm. diameter. In some instances, however, a coarse-grained structure was noted, occa-

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sionally as large as 0.40 mm. diameter, usually in the heavier sizes. Sheet is, of course, subject to rapid cooling during the rolling operation and the presence of coarse grains in the finished material is probably due to the temperature falling within the critical range before rolling is completed, grain-growth occurring during cooling in the stock pile or on annealing. No grain-growth was observed in the heavier materials, such as hot-rolled rod, plate, or sheet bar, either after rolling or after annealing within the critical range. Further annealing within the critical range seemed to have no effect on the grain-size of the sheet investigated.

The grain-size noted in cold-rolled annealed sheet varied considerably, but where grain-growth was found, the size was generally coarser than that found in hot-rolled. Grain-growth occurred in all sizes down to and including 0.040 in., the lightest gage examined. Grains 0.50 mm. in diameter were noted occasionally.

In many cases where sheet is used for such operations as pressing, a considerable increase in the number of operations becomes necessary when the material is coarse-grained; frequently the grain-size may be so large that the steel will not stand even light operations. In such cases the material can be made available for use by annealing at about 790 deg. C. (1450 deg. F.) or in some cases, above 900 deg. C. (1650 deg. F.). When the operations have been laid out with normal metal, the adjustment that may be necessary to handle the coarse-grained material may cause considerable difficulty.

Detecting and Eliminating Grain-Growth

There is no convenient method by which the condition of sheet can be determined absolutely. Microscopic examination together with a tensile test will generally indicate to a certain degree the condition of the metal and its probable action in practice. These tests cannot be an infallible guide because the strain to be applied may be too complex, or because the coarse structure may occur only here and there throughout the metal in certain areas which may or may not be found by the test. Moderately coarse-grained sheet that gives perfect satisfaction in one case may be very unsatisfactory in another. In general, however, the test stated will indicate material of decidedly coarse grain and also material that has not been handled properly in rolling or annealing. A few positive results of this kind will soon pay for any negative results that may be obtained. If the sheets have all been rolled at the same time, samples taken from a small percentage will usually be representative of the entire lot. These methods of examination are applicable to other forms of cold-worked low-carbon steel.

The problem of eliminating grain-growth in sheet is rather more difficult to solve than that of detecting it. With hot-rolled sheet prevention requires careful regulation of the finishing temperature, the reduction and the annealing. When thin sheet is rolled, such regula-

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tion is exceedingly difficult. With cold-rolled sheet grain-growth can be eliminated by heavy drafts, in the same way that strip is rolled, or by regulation of the annealing temperature. Surface irregularities in the hot-rolled sheet and the difficulty of measuring the gage exactly with large thin sheets make an absolute determination of the reduction exceedingly difficult. Furthermore, it is not at all improbable that in some cases a coarse structure may persist from the hot-rolling. To insure the elimination of coarse grains in sheet, the annealing of that hot-rolled above 900 deg. C. (1650 deg. F.) is required, and if then cold-rolled the temperature of the subsequent annealing should be kept below 650 deg. C. (1200 deg. F.). Sheet treated in this manner will be somewhat stiffer than usual, but can generally be strained to a considerable degree before annealing becomes necessary. The difficulty of annealing sheet at the high temperature is obvious, but box-annealing should be feasible.

Grain-growth in cold-rolled strip steel can be prevented by the use of reductions of area above 25 to 30 per cent. The gage can be determined accurately, permitting close control of the reduction in rolling.

Growth in Annealed Tube

Examination of a number of samples of cold-drawn annealed tube frequently showed moderate grain-growth, although not of a serious nature, as in no case were any grains over 0.15 mm. in diameter found. Heating to 900 deg. C. (1650 deg. F.) was necessary in order to refine the structure.

Cold Pressings

In certain cold pressings grain-growth may readily occur and cause serious difficulty in the process of manufacture or in use. Where 0.20 per cent carbon steel is used there is practically no danger of grain-growth, but in the usual practice low-carbon steel is used, in which grain-growth can readily be produced. The occurrence of grain-growth in pressings is dependent upon the nature of the operation and the annealing temperature. The annealing of pressings is a process that is frequently looked upon as requiring elementary control, the only consideration being that the metal reach a temperature high enough to soften it. The usual tendency is to keep the temperature high enough to insure this softening, and so it may frequently rise within the critical range where grain-growth occurs. A large number of pressing operations are light, and improper annealing may cause extensive grain-growth. Control of the annealing temperature is of the utmost importance in these cases.

The selection of the temperature depends to some extent upon the nature of the operation and the condition of the metal. Although annealing above 900 deg. C. (1650 deg. F.) or below 650 deg. C. (1200 deg. F.) will produce no grain-growth, the use of these

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temperatures is not always feasible. Annealing just below 690 deg. C. (1275 deg. F.) will usually produce grain-growth so slight that it will not be the cause of any trouble in further operations. This temperature will remove practically all the hardness resulting from cold work, although in a few cases a higher temperature may be required. Certain upsetting operations (7) require annealing at a temperature somewhat above 690 deg. C. (1275 deg. F.) and there are a few pressing operations of a similar nature. In such cases or where a decidedly coarse-grained structure exists in the metal, annealing it at about 790-800 deg. C. (1450-1470 deg. F.) may refine the grain sufficiently; if not, 900 deg. C. (1650 deg. F.) will be required.

Grain-growth of any extent can occur only when the applied strain has been comparatively light, and control of this factor may prevent its occurrence. Increase in the extent or number of operations before annealing will generally permit the use of almost any normal annealing temperature. Very often, however, the effect of the operation cannot be foretold and occasionally some variation from normal may occur, as for instance, in the setting of the punch or die, resulting in a comparatively light strain in some unexpected place. Furthermore, there are many operations which must necessarily be very light, and where annealing is required, its proper regulation becomes essential.

Cold-drawn Rivet Steel

Another form of low-carbon steel in which grain-growth may occur is the cold-drawn rivet. When such material is intended for cold heading, care is taken to insure its being in the best possible condition and free from coarse crystallization. When intended for hot work, such care is not essential. Trouble may develop if rivets of the latter type are used for cold work, especially if annealed within the critical range. As a general rule rivets requiring annealing should be heated above 900 deg. C. (1650 deg. F.).

LOW-CARBON STEEL PRESSINGS IN MOTOR VEHICLES

The low-carbon steel pressing plays an important part in the design of motor vehicles. Such parts as frames, hubs, housings, fender and running-board supports, pans, casings, flywheels, clutch cones, roller-bearing cages, ball cups and races are usually low-carbon steel pressings, in which grain-growth may develop under improper treatment. As it is in the earlier operations that grain-growth usually appears, making further work on the material impossible until it has been properly annealed, only a comparatively small proportion of the product will find its way into service in a coarse-grained condition. In some cases, however, the presence of coarse crystallization may not be detected by the maker and defective

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parts may find their way into service. The cause of the trouble in many such cases may be obscure as, for instance, in the occurrence of cracks along the blanked edges of pressings. This is generally due primarily to the pressure of the cutting edge in the blanking operation, subsequent annealing producing a moderately coarse-grained condition in a narrow layer following the line of pressure.

The presence of this coarse-grained condition may not be detected in the manufacturing operations, or perhaps the annealing may be a final operation, so that no trouble is developed until the parts are placed in service. After a short time cracks appear along the edge and extend rapidly through the section. In a general way this condition has been observed by manufacturers of pressings, who occasionally find it necessary to grind the edges of certain parts in order to continue the operations without developing cracks. This same condition may occur around punched holes. Trouble in service from this source will usually be confined to certain parts, the edges of which may be subjected to considerable bending moment, as for example, frames, running-board supports and fender brackets.

Pressed ball-cups and races should be heated to above 900 deg. C. (1650 deg. F.) during the carbonizing process to remove any previous coarse crystallization. Distinction should be made here between the grain-growth following the pressing operation and the grain-growth which may be developed from the use of a high temperature in the carbonizing operation.

GENERAL METHODS OF PREVENTION AND CURE

Many operations on low-carbon steel, especially in pressing, are of such a nature that no annealing is required, and when eliminated no grain-growth can occur. But whenever annealing is essential, the temperature must be carefully selected and controlled. These considerations must be borne in mind by the user as well as the maker, for it may become necessary to apply further heat treatment to some material before assembly, and improper treatment may leave the material in a dangerous condition.

Where grain-growth has occurred and refining becomes necessary, heating above 900 deg. C. (1650 deg. F.) will be found to refine the grain completely. If the scale produced at this high temperature should be troublesome, the material may be quenched in water, subsequently annealing at 540 to 675 deg. C. (1000 to 1250 deg. F.) if necessary. A little concentrated hydrochloric acid thrown into the furnace at the start will loosen the scale so that it can be readily removed. In some cases annealing at 790 deg. C. (1435 deg. F.) will refine the grain sufficiently to prevent further trouble.

To summarize the foregoing, it may be stated that grain-growth in cold-worked low-carbon steel can be avoided by the use of comparatively heavy operations on the metal before annealing or by

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proper selection and control of the annealing temperature, according to the working conditions. If the material is not annealed, no grain-growth can occur.

AUTHOR'S CONCLUSION

1. Grain-growth in low-carbon steel may be produced by permanent deformation within certain limits, followed by annealing within certain temperature ranges.

2. The greater the amount of strain applied within these deformation limits, the smaller will be the grain-size produced by annealing.

3. The annealing range within which grain-growth due to strain beyond the elastic limit may be produced, usually lies between 650 and 900 deg. C. (1200 and 1650 deg. F.).

4. When the applied strain falls below a certain, or "critical," amount, the annealing range within which grain-growth can occur seems to be limited by the thermal critical points at about 690 and 780 deg. C. (1275 and 1435 deg. F.)

5. The most practical standard by which to measure the applied strain seems to be the reduction of area. Experiments based upon such measurements show that no grain-growth occurs following a reduction of area of less than about 7 or more than about 25 to 30 per cent. The "critical strain" mentioned seems to be marked by a reduction of area of about 9 per cent.

6. The refining action taking place at about 780 deg. C. (1435 deg. F.) is a further indication of the presence of a thermal critical point at this temperature.

7. No serious grain-growth was found in steel the carbon content of which was uniformly above about 15 per cent.

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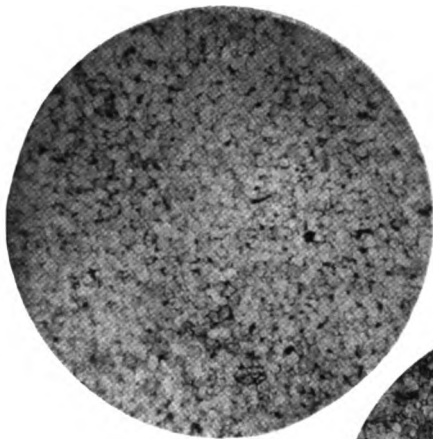
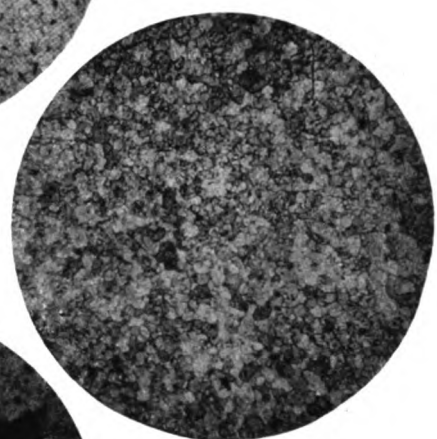
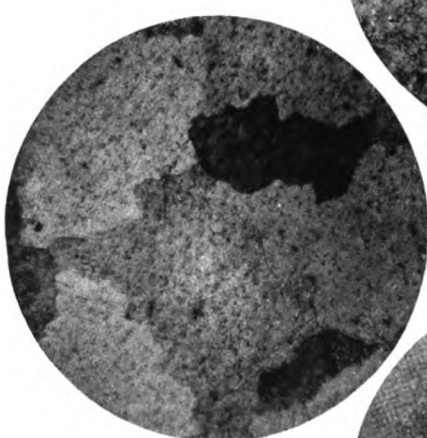


Original Rod Annealed One Hour at 700 Deg. C.



Rod After Drawing, Annealed One Hour at 675 Deg. C.



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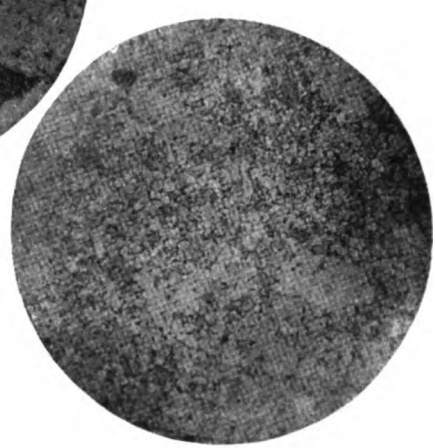

Rod After Drawing, Annealed One Hour at 780 Deg. C.

PLATE I—RESULT OF ANNEALING ROD DRAWN ABOUT 9 PER CENT REDUCTION OF AREA
Magnification—50 Diameters

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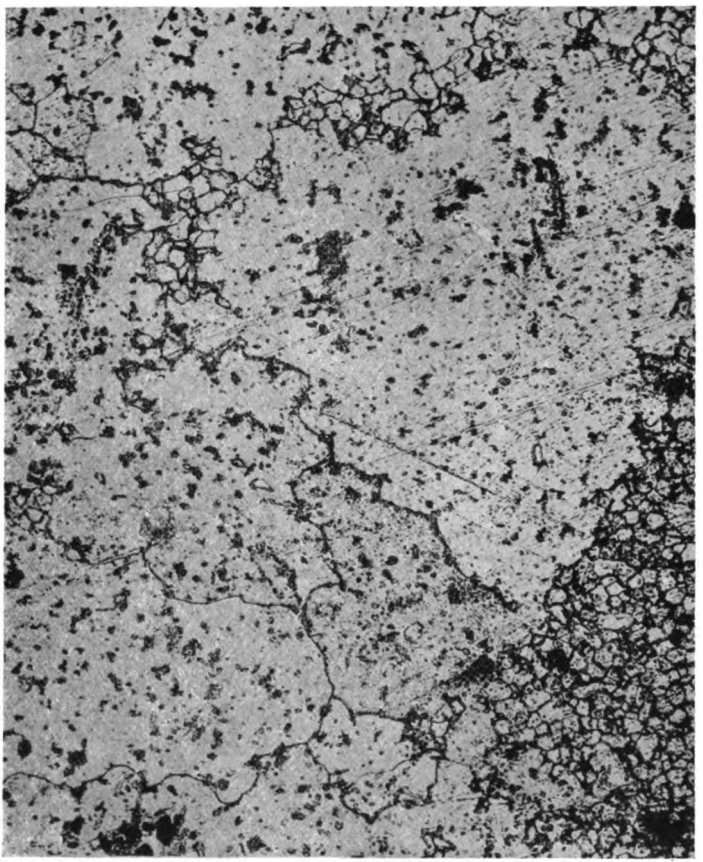

Annealed one hour at 700 (above) and at 790 (below) Deg. C.

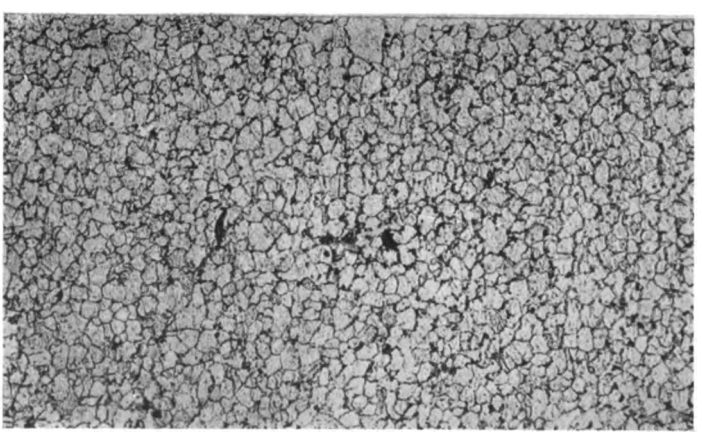

PLATE II—ROD 0.320 IN. DIA. DRAWN 8 PER CENT REDUCTION
Magnification—100 Dia. Reduced About 10 Per Cent in Reproduction

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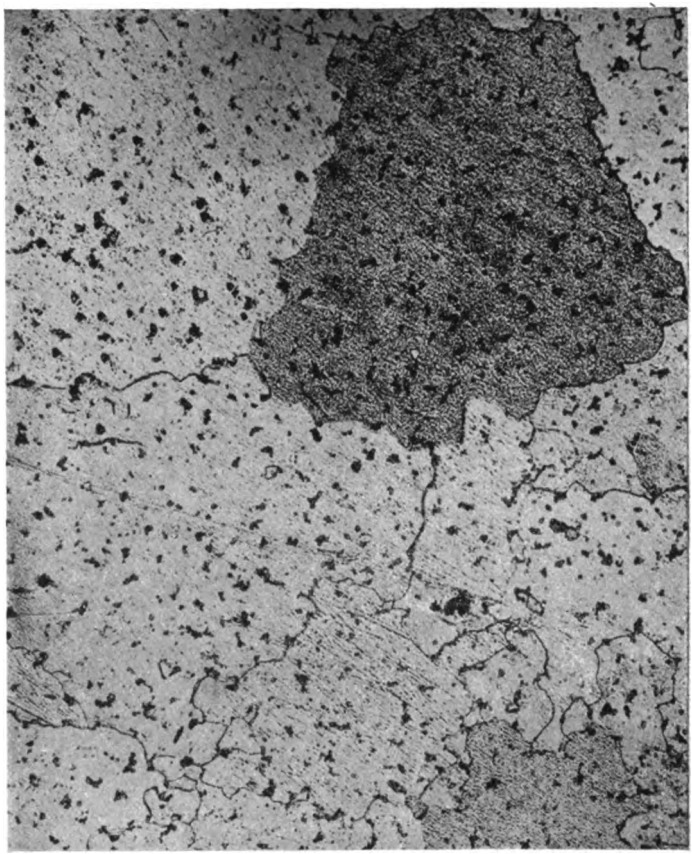

Annealed one hour at 700 (above) and at 790 (below) Deg. C.

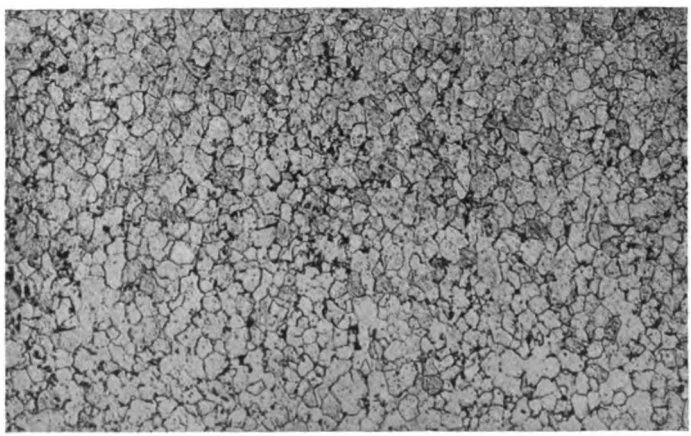

PLATE III—ROD 0.320 IN. DIA. DRAWN 9 PER CENT REDUCTION
Magnification—100 Dia. Reduced About 10 Per Cent in Reproduction

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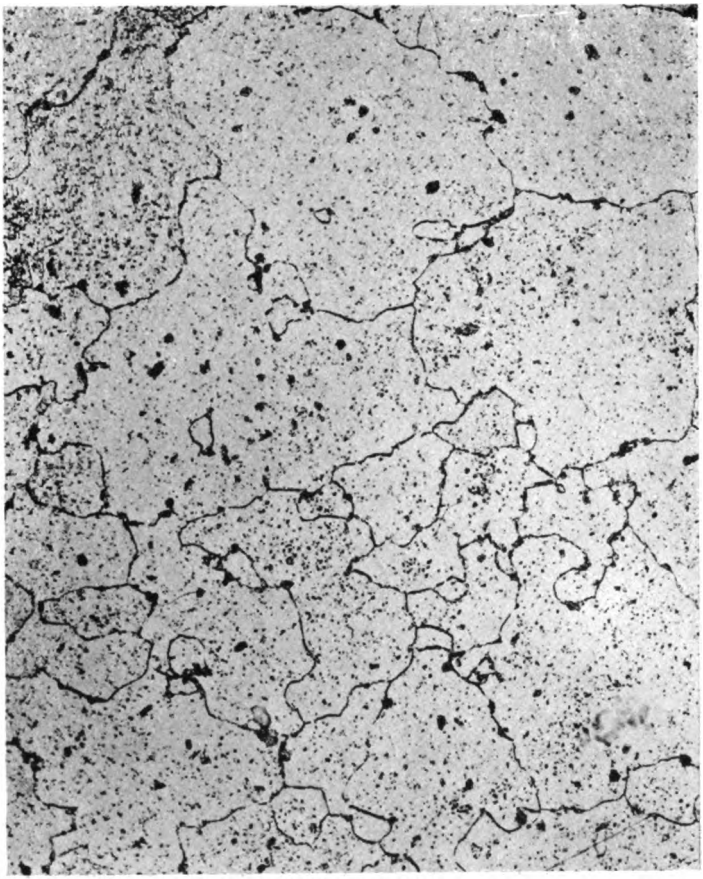

Annealed one hour at 700 (above) and 790 (below) Deg. C.

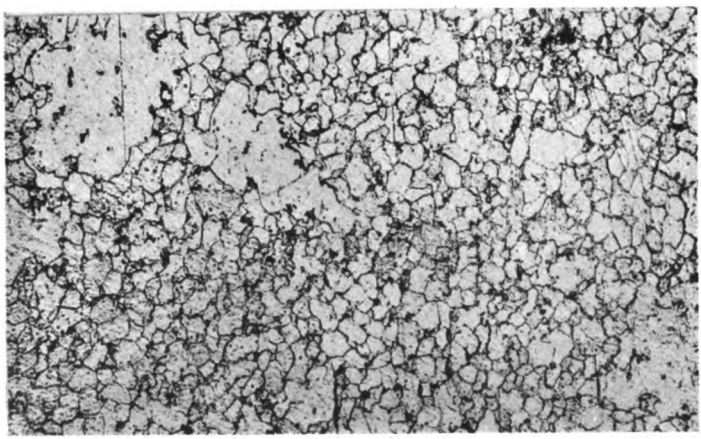

PLATE IV—ROD 0.320 IN. DIA. DRAWN 10 PER CENT REDUCTION
Magnification—100 Dia. Reduced About 10 Per Cent in Reproduction

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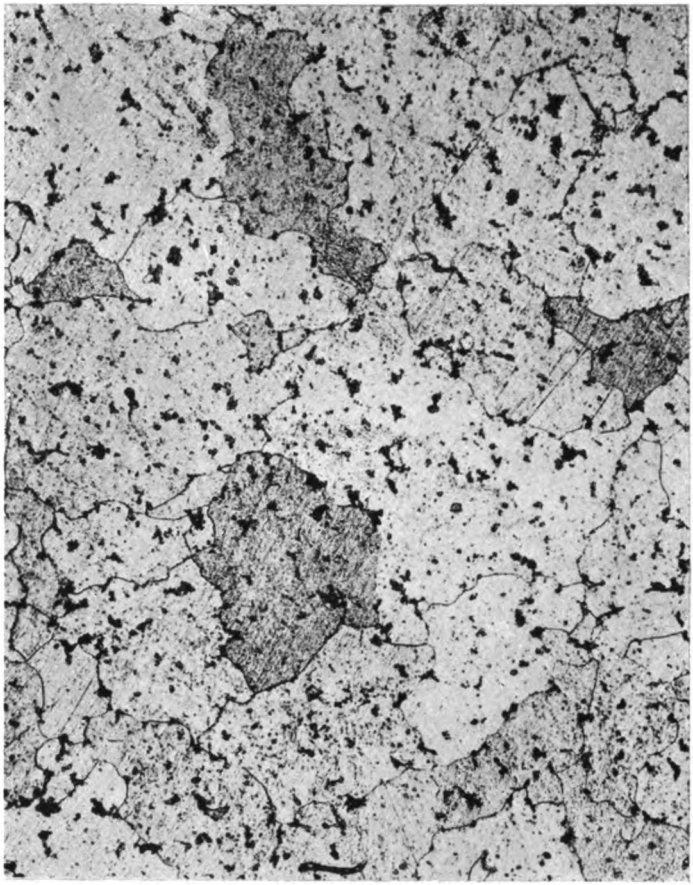

Annealed one hour at 700 (above) and 790 (below) Deg. C.

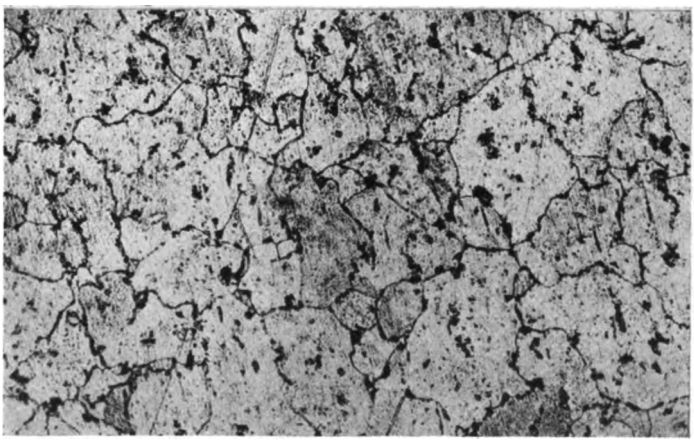

PLATE V—ROD 0.320 IN. DIA. DRAWN 11 PER CENT REDUCTION
Magnification—100 Dia. Reduced About 10 Per Cent in Reproduction

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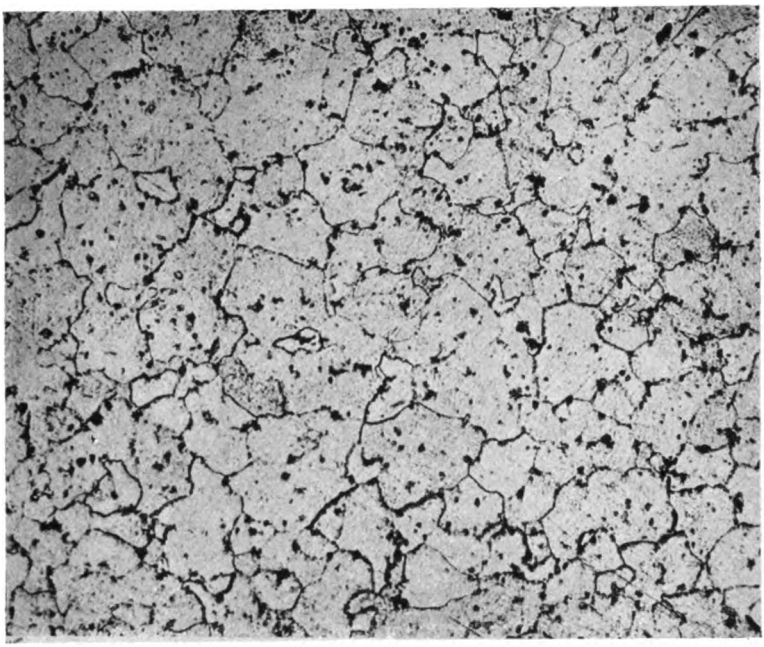

Drawn 12.5 per cent (above) and 16 per cent (below) Reduction

PLATE VI—ROD 0.320 IN. DIA. ANNEALED ONE HOUR AT 700 DEG. C.
Magnification—100 Dia. Reduced About 20 Per Cent in Reproduction

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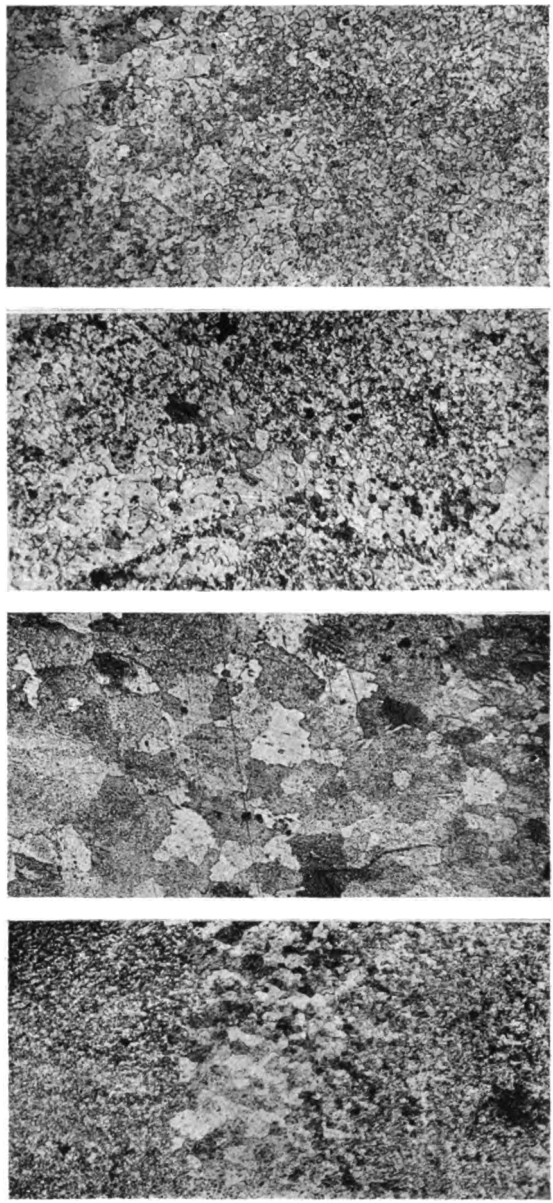
PLATE VII—PARALLEL STRIPS FROM PRESSING. VIEWED FROM BOTTOM OF PRESSING. ANNEALED 30 MINUTES
Magnification—250 Dia. Reduced About 40 Per Cent in Reproduction



No. 1—Annealed at 675 Deg. C. No. 2—Annealed at 705 Deg. C. No. 3—Annealed at 790 Deg. C. No. 4—Reheated at 790 Deg. C.

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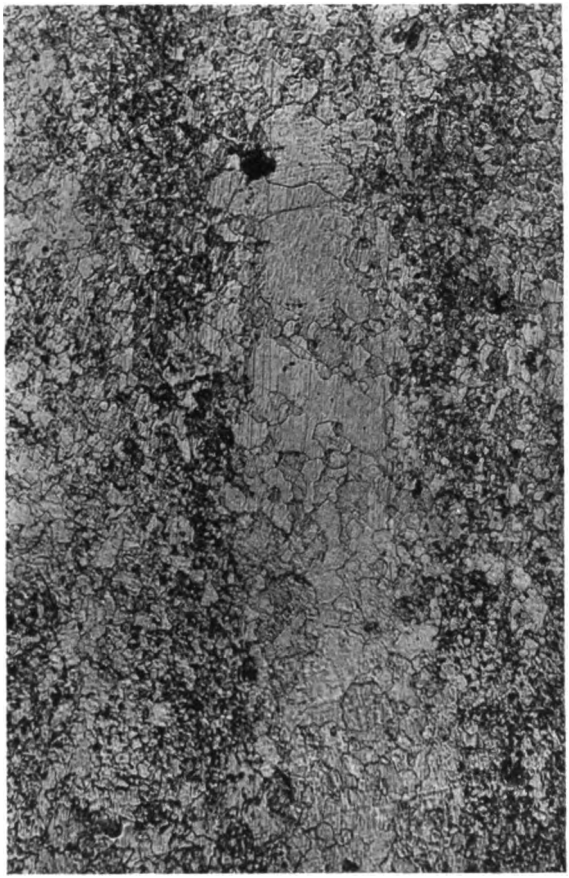

PLATE VIII—GRAIN GROWTH IN FERRITE BAND OF HOT-ROLLED 0.18 PER CENT CARBON STEEL.
Magnification—100 *Dia.*

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BRONZE ALLOYS IN AUTOMOBILE CONSTRUCTION

W. M. Corse and G. F. Comstock
(Member of Society) (Non-Member)

Abstract

The authors point out the need for more concrete data concerning the physical properties of bronze alloys and present an extensive chart covering the results of actual tests on a large range of cast bronze alloys. The influence of the method of making the test-specimen is discussed and it is hinted that new evidence concerning the proper interpreting of the true proportional limit is available.

A very extensive set of microphotographs illustrates the variety of structures existing in the ordinary and in some unusual bronze alloys.

ALLOY DATA NEEDED

Experience in the bronze casting business has demonstrated the need of many data on the physical and chemical properties of alloys that are not available in handbooks or other sources of information on the subject. Tables showing such properties determined from actual practice, which may be termed experience tables, are rare; some data exist but they are usually in the private files of individuals or companies. The Bureau of Standards at Washington reports daily requests for such information and states that there is great difficulty in obtaining data so that correct replies can be made.

With this need in mind the committee on non-ferrous alloys of the American Society for Testing Materials discussed at a recent meeting the advisability of publishing such data as might be compiled from existing experience tables. On account of the fact that publication of data by that society is made in a formal manner, usually as specifications, it was thought best to delay so as to allow the experience tables to be checked by further work. The need was fully realized, however, and it is with that idea in mind that the table included in this paper was compiled.

One very important point should be emphasized in connection with the use of the table mentioned. Most tests are made on standard test bars either attached to castings or cast separately. Test bars cut from castings, where this procedure is possible, show conclusively that a variation exists between results on standard test bars and the metal in the castings themselves.

Castings may show higher or lower results, depending on the design and size of the casting itself. The latter is usually the case

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SPECIFICATIONS AND RESULTS OF TESTS OF

Name of Alloy	Composition								Tensile		
	Cu.	Sn.	Zn.	Pb.	Al.	Fe.	Mn.	P.	Yield Point, Thousands of Pounds per Square Inch	Elastic Limit, Thousands of Pounds per Square Inch	Ultimate Strength, Thousands of Pounds per Square Inch
Bronze	80	20						tr.			
Grade "A" bridge bronze	80	20						1 (a)			
Special phosphor bronze	80	20						0.2 (i)			
Grade "B" bridge bronze	85	15						1 (a)			
Special phosphor bronze	99							1 (a)			
Hard bronze	82.5–83.5	13.8–14.2	1.1–2.9	0.8–1.2						15–19	25–27
Gear bronze	85	13	2						23–25		27–30
Phosphor bronze	79.5	12		7.5				1			30–34
Tough bearing bronze	85.5–86.5	10.5–12	1.5–3.5	1(a)		0.25 (a)	0.15 (a)			15–20	34–32
Stone's English gear bronze	89	11							22–25		26–40
Gear bronze	90	10.5						0.5			31–35
Cast gun metal, "G," U. S. N.	87–89	9–11	1–3	0.2		0.06 (a)			15 (i)		30 (i)
Tough bronze	85.7–89.3	9–11	1.5–2.5	0.3–0.8						13–16	28–34
Gun metal bronze	86–89	8–11	1–3								33 (i)
Bronze	88	10	2								33–36
Grade "D" bridge bronze	88	10	2					0.25 (a)			25
Gun metal	88	10	2						19–22		33–36
Bronze 3, U. S. N.	87–91	8 (i)	4 (a)	1 (a)		0.5 (a)		0.5 (a)			30 (i)
Cast phosphor bronze, Pt. Grade 1, U. S. N.	88–90	9–11	4 (a)	0.2 (a)		0.06 (a)		0.5 (a)			45 (i)
Cast phosphor bronze, Pt. Grade 2, U. S. N.	88–90	9–11	Remainder	1 (a)		0.1 (a)		0.5 (a)			30 (i)
Bronze	90	10							22–34		28–45
Bronze	90	10						tr.			33–36
Bronze	89.5	10			(0.5 Sb)		(Included in Cu)				
Bronze	89	10		1							23
Bronze	89	10		2					19–23		30–36
Bronze	79.7	9–11		8–11				0.7–1			
Phosphor bronze	89	10		10					20		20
Phosphor bronze	79.75	10		10				0.25			28–32
Grade "C" bridge bronze	89	10		10				0.7–1			
Bronze	89	10		10				0.8			
Phosphor bronze	89	10		10					19–21		28–32
Bronze	89	9.2		10				0.8			
Bronze	84	9.5		6.3				0.2			29–33
Bronze	89	8	2	2							31–37
Free-machining bronze	87.5–89.5	7–7.5	0.4–3.4	1.5–2.3						11.5–15.5	21–31
Bronze	81	7	3	9					19–20		30–36
Bronze	78	7		15					17–18		32–31
Bronze	90	6.5	2	1.5							34–40
Bronze	90	6.5	2	1.5					16–22		34–40
Bronze	85	5	5	5							27–33
Red composition, ounce metal	85	5	5	5					15–19		27–33
Bronze	80	5	2	3					16		34
Red brass	85	4	7	4					14–16		28–33
Red brass	85	3	9	3							23–26
Red brass	82	2	5	10					10–11		19–28
Bronze, 2, U. S. N.	65–70	2 (a)	25–27	1 (a)		0.5 (a)					25 (i)
Natural color brass	74.5–77.5	0.5–1.2	21–23.5	0.3–1.2		0.3 (a)				7–11	23–28
Yellow brass	70	1	27	2					10–11		23–31
Yellow brass	69	1	30								28–32
Bronze, 1, U. S. N.	55–62	1.5 (a)	Remainder	0.4 (a)	1.5 (a)	0.5–2	0.5–2.5				60 (i)
Cast manganese bronze, Mn-C, U. S. N.	55–60	1.5 (a)	38–42	0.2 (a)	1.5 (a)	2 (a)	3.5 (a)				65 (i)
Manganese bronze	57.75–58.25	0.5–0.5	38.5–39.5	tr.–0.2	0.5 (a)	1.05–1.35	tr.–0.25			26–34	55–70
Manganese bronze	56	0.75	41.5		0.5	1	0.25				72–84
Manganese bronze	57	1	40		0.75	1	0.1			27 (i)	75
Manganese bronze	56	1	41		0.5	1	0.5		33		70
Manganese bronze	56–57	1	38–40			1.5					
Manganese bronze	53–62		38–45	0.15 (a)	0.05–0.5						70 (i)
Manganese bronze	60		40						30		60
Yellow brass	56		45								30–45
Vanadium bronze, Va-C, U. S. N.	61 (i)	1 (a)	38 (a)						23.5 (i)		55 (i)
Tenaflite	64	0.2	29		3.1	1.2	2.5		30		98
Tenaflite	67		24		4.4		3.8	0.01			
Tinaloy aluminum bronze	90				10				23–25		70
Aluminum bronze	90				9.9				25		71
Tinaloy aluminum bronze	89				10	1			24–30		70
							Ni				
Cast monel metal, Mo-C, U. S. N.	Remainder			0.0	0.5	0.5 (a)	60 (i)		32.5 (i)		65 (i)
Monel metal	27					6	67		37		75

NOTES: (a) Maximum limit. (i)—Minimum limit. Other figures show either approximate average or range. Compiled from

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COMMERCIAL BRONZE AND BRASS CASTINGS

Test		Hardness		Compression Test		
Elongation in 2 Inches, per cent	Reduction of Area, per cent	Brinell	Scleroscope	Elastic Limit, Thousands of Pounds per Square Inch	Per Cent Reduction in Height After Loading to 100,000 Pounds per Square Inch	Uses, Remarks, Etc.
		136			3 1	Dies, bolts, bearings, etc.
				25–40	6–10	In bridges, for contact with hard steel under pressure above 1500 lb. per sq. in.
				34 (i)		Disks for draw bridges.
				19–22	12–25	In bridges, for contact with soft steel at pressure below 1500 lb. per sq. in.
				19–22	12–16	Trunnion bearings.
tr.–4	tr.–4		20–24			Bearings and bushings.
2–3	1–4					Gears.
1–2		74–80				Bushings and bearings not subjected to shock.
2–15	2–15		15–22			Bearings, gears, etc.
6–10	7–8	80		15.5–16 5	25–27	Gears and worm wheels.
6–10		72–77				Worm, spur and bevel gears.
15 (i)						Valves, gears, etc., and where strength and incorrodibility are desired.
10–22	9–21		14–18			Durable castings.
14 (i)						(A S.T.M. tentative specifications.)
14–18		70–74				Hard bearings, hydraulic work, valves, machine parts, etc.
14				14 (i)		Gears, worm-wheels, nuts, etc.
14–18	12–15	70–75		14 5–15	27–29	Bearings for heavy pressures and high speeds, thrust collars, steam fittings, hydraulic work.
15 (i)						Water-tightness and bearing qualities in thin sections.
20 (i)						Castings for strength, incorrodibility and good bearing qualities.
15 (i)						Less severe service than Grade 1.
15–18	15–20	65–70		15 5–16	26–29	To resist acid, and for thrust collars and disks under high pressure.
10–14		67–70				Pumps, collars, bearings, and to resist corrosion.
				15		Worm wheel rims (copper content includes manganese).
				23		Worm gearing, safe working stress: 7000 lb. per sq. in.
15–20	20–26	65–70		15	26–29	Gears under moderate service, and for good machinability.
				15–20		Bushings and shaft bearings.
6						(S. A. E. Specifications.)
5–7		65–70				Bearings for high speeds and heavy pressures.
				15–20		Ordinary machinery bearings.
				15		Bridge bearings.
5–7	6–10	55–60		11 5–12	32–35	Standard for high speeds, heavy pressures, shock and vibration.
				27	6 25	
5–10		59–61				Slide valves, bearings, and for resisting acids.
16–20		57–60				Bearings, small valves, steam fittings.
11–15	7–13		8–12			Gears and pump castings.
15–18	22–24	50–55		12 5–13	34–35	Inexpensive bearings.
12–18	17–23	48		11–12	36–39	Bearings for medium pressure and high speed.
25–33		54–57				Bearings, steam fittings, and for incorrodibility.
25–33	26–34	50–60		12 5–13	33–34	Small automobile bearings, or for high speed and heavy pressure, and steam fittings
16–20		52–53				Valves and bearings.
16–20	15–20	50–60		11 5–12	26	Pump bodies, valves, steam fittings, and bearings.
26–31	29	42–45		11 5	34–35	Valves, steam fittings and metal patterns.
15–20	20–26	55–60		9 5–10	37–39	Valves and pipe fittings.
10–16						Oil and water pumps, and trolley fittings.
7–13	11–17	34		7–8	46–47	Inexpensive bearings for low pressure and no shocks, valves, fittings, etc.
15 (i)						Miscellaneous fittings, not subjected to friction, corrosion or pressure.
23–33	15–25		6.5–9.5			Oil feed, cylinder top cover, etc., for automobiles.
22–26	27–30	50–65		6	40	Light castings and ornamental work not requiring strength.
22–26		57–60				Ornamental work.
15 (i)						For strength in heavy work, substitute for steel in intricate designs, resistance to corrosion.
20 (i)						Castings requiring great strength.
18–30	22–30		17–23			Strong castings.
22–25		109–119				Propellers, engine brass, and all strong castings.
16	22					Strong castings.
22–25	26	104–119		27 5–28.5	10–11	Propellers, valve stems, engine framing, and castings requiring strength and toughness.
				25–40		
20 (i)						Ingots for castings (A.S.T.M. specifications).
20						(S. A. E. Specifications.)
9–13		50–65				Low grade bearings.
25 (i)						(Tin content includes all other impurities.)
16	17	130				(Single test.)
				60	1.33	Railroad turntable.
20	21	90–100		19–20	11.4	For strength, toughness and especially incorrodibility.
22						(Eighth Report to Alloys Research Committee, British Institute of Mechanical Engineers.)
24.5	27.3	90–96		18–20	12–15	For strength and resistance to wear, pressure, or repeated shocks.
25 (i)						Fittings, etc., and castings requiring great strength.
26						

16 different sources, but in each case the physical data come from the same authority as the corresponding chemical composition.

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on account of the fact that the average size of castings exceeds the test bar size. It has been proved conclusively that the rate of cooling and the shape of the casting affect the results markedly.

Alloys of high shrinkage are much more liable to show internal strains due to practical inability to feed the shrinkage properly, so that the net strength is the difference between the true value for the metal and the internal strains caused by this shrinkage. Manganese bronze for example normally shows an average tensile strength of from 65,000 to 75,000 lb. per sq. in. It is not uncommon to find this lowered to 30,000 or 40,000 lb. per sq. in. in actual castings.

Copper tin alloys such as phosphor-bronzes do not exhibit high shrinkage but show the same proportional discrepancy in strength, on account of large crystal growth and the formation of a coarse network of the high tin eutectoid, which is very hard and brittle. Thurston, for instance, in his standard work on alloys*, quotes from Major Wade's report of 1856 on "Experiments on Metals for Cannon," in regard to gun bronze containing 10 per cent tin, that 83 test bars from gun heads of castings averaged about 30,000 lb. per sq. in. tensile strength, while 32 small bars averaged about 42,000. Also, "The average of 12 gun heads was one-half (in strength) of that obtained from small sample bars cast with the guns." The area of the cross-section of the casting, as well as its design therefore, has a very important bearing on the subject and should always be taken into account in engineering work.

Improper foundry practice affects the tests of any alloy to a very marked extent, so that two metals of the same chemical analysis can show very different results when handled by different foundries. The latter statement is almost self-evident but many engineers do not realize the effect of small amounts of impurities, some of which are acquired during the melting process, on the final result.

We may learn two things from the above. First, to secure if possible the maximum amount of information on castings themselves even at the expense of a few spoiled pieces. Second, to employ the most reliable foundryman obtainable, as the best practice is none too good with the appliances existing in the foundry.

VALUE OF MICROSCOPE

To the above might be added a suggestion to examine the structure of the metal with the microscope. We all know the value of such a procedure with steel, but so little work has been done with the bronzes that its value here may easily be overlooked. As a preliminary step in this direction a number of microphotographs are included in this paper. In similar pieces grain-size can be easily compared, as in Figs. 19 and 22, which are taken from the same composi-

*"A Treatise on Brasses, Bronzes, and Other Alloys, and Their Constituent Metals," by R. H. Thurston, Fourth Edition, 1910, Part 3, Pages 306-307.

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tion, one being made in a sand mold and the other in a chill mold. In Figs. 1 and 2 we see different structures. The chemical compositions are identical but the copper tin eutectoid is clearly seen to be present in Fig. 2 and almost absent in Fig. 1.

The high copper aluminum alloys commonly known as aluminum bronzes are very interesting from a metallographic standpoint. In Figs. 20 and 21 we have two metals of the same composition but with different structures, due to "self-annealing" or heat treatment in the mold during the casting process. Tests show a difference in tensile strength of 11,000 lb. per sq. in. and in elongation of 10 to 12 per cent. The microscope was a distinct help in explaining these differences. Our experience in the last two years indicates increasingly the value of metallography as applied to bronzes.

Many engineers believe that when figuring loads it is not safe to go above the value of the true proportional limit as shown by the extensometer readings in the tension test. An interesting point has arisen recently in connection with aluminum bronze. The fatigue test as made with several types of machines indicates clearly that the primitive yield point as shown by the tension test is not the true yield point of the material. The endurance or fatigue resistance of this alloy exceeds that of manganese bronze, or even of steel under certain conditions, although the primitive yield point of manganese bronze is above that of aluminum bronze. It would seem therefore that valuable information might be obtained by considering the fatigue resistance of a material in addition to the usual properties shown in the table. Further information on the subject can be obtained by referring to a paper presented before the American Society for Testing Materials at its 1916 meeting.*

MATERIALS FOR BEARINGS

We have avoided specific statements or recommendations about materials for bearings because it is difficult to predict the result with several other variables undetermined. For instance, the hardness of the steel shaft and its machine finish are as much a factor in the selection of the bronze as the properties of the bronze itself. Moreover, the quality of the surface finish needed on the bronze bearing is also determined by the ability of the bronze to conform to the steel. It is evident that as the bearing pressure increases the hardness also must increase to avoid flowing. In order to have a satisfactory hard bearing it is necessary to have a finely machine-finished surface on both bearing and shaft because they will not conform as will softer metals.

Inasmuch as there are many different bearing conditions of pressure and speed, it can be seen that no one bronze can fill all the needs of the automobile engineer. In a general way tin and zinc harden bronze and lead softens it. The use of zinc in small proportions is

*"Aluminum Bronze; Some Recent Tests and Their Significance," by W. M. Corse and G. F. Comstock, Proc. A.S.T.M., June 1916.

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principally for deoxidizing purposes, but has a detrimental effect on the wearing properties. Consequently, when conditions require hard bronze bearings, the tin content must be relatively high. It sometimes goes to 15 per cent in automobile work.

Lead also may go to 30 or even 40 per cent for special work, such as in racing cars, but its effect should be considered in relation to the tin content as well as by itself. It is a good lubricant in bronze. Present practice frequently calls for a combination bearing with a relatively hard back and a soft lining of babbitt metal. The selection of babbitt should be determined by the thickness of the lining, and pure tin or solder can be used if the lining is very thin.

SUMMARY

In order to aid engineers in choosing the right alloy, we must first recognize several facts.

First—Standard test bar results do not necessarily represent the material in the castings. Wherever possible, secure tests from pieces cut from castings as a check on the properties the metal is shown to possess from test bar results.

Second—The aid of competent foundrymen should be secured in making castings as the chemical analysis or standard test bar results do not tell the whole story.

Third—The microscope is as important in determining the properties of bronze as those of steel. More information on the subject is needed.

Fourth—The endurance or fatigue test is an important aid in determining the quality of a bronze and like the metallographic test should be used to be appreciated.

Fifth—The proper composition to use in automobile work depends upon the kind of service required. For conditions requiring hardness, high tin is needed. If conformability is more essental, the lead content should be high.

Sixth—Published data are very meagre, so that any additional reliable information will be a distinct gain to the literature of the subject.

If the facts given arouse the interest of automobile engineers and stimulate their desire to know more of the subject of bronzes, the purpose of this paper will have been accomplished.

EXPLANATIONS OF ILLUSTRATIONS

Fig. 1—Structure of chilled face of phosphor bronze gear, containing 11 per cent of tin, magnified 200 diameters, and etched with ammonia and hydrogen peroxide, followed by ferric chloride. The metal is practically all alpha crystals, with only a few specks of the bright eutectoid. The dark spots are the soft cores of the alpha crystals.

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Fig. 2—Same alloy as shown in Fig. 1, but cast in sand. Etched and magnified like Fig. 1. The slower cooling has produced a coarse grain and allowed large amounts of the eutectoid to separate between the alpha crystals.

Fig. 3—A bronze containing 13 per cent of tin and 2 per cent of zinc. Otherwise like Fig. 2. There is more eutectoid here on account of the higher tin content.

Fig. 4—Section of a hard spot on the surface of a phosphor bronze gear containing 11 per cent of tin, magnified 200 diameters, and etched with ammonia and hydrogen peroxide, followed by ferric chloride. This material is composed largely of eutectoid, with dark grains of alpha solid solution. The spot was formed by liquidation of the more fusible high tin constituent into drops at the surface of the casting while it was cooling in the mold.

Fig. 5—View of the hard spot shown in Fig. 4, but magnified 20 diameters. The bright portion of the microphotograph is the hard spot, bounded above by space beyond the edge of the gear, and below by the dark etched normal metal, like that shown in Fig. 1.

Fig. 6—Gun bronze, 88 per cent copper, 10 per cent tin, 2 per cent zinc, cast in a chill mold, etched with ammonia and hydrogen peroxide and magnified 20 diameters. This etching darkens only the soft cores of the alpha crystals, whose dendritic structure is here well shown.

Fig. 7—Same as Fig. 6, but magnified 200 diameters. The soft cores are most prominent here, but small indistinct particles of eutectoid are also present in the bright areas.

Fig. 8—Structure of English worm gear containing about 89 per cent copper, 10 per cent tin and 0.5 per cent each of zinc and lead, etched and magnified like Fig. 6. The much coarser grain was caused by slow cooling.

Fig. 9—Same as Fig. 8, but magnified 200 diameters. The dark spots are the soft cores of the alpha crystals. The irregular spotted constituent is the hard brittle eutectoid, which formed abundantly in large pieces on account of slow cooling. This metal would be harder to machine than metal illustrated by Fig. 7 or Fig. 1.

Fig. 10—Structure close to surface of bearing of 80 per cent copper, 10 per cent tin and 10 per cent lead. Etched with ammonia and hydrogen peroxide and magnified 20 diameters. Some of these darker spots are soft cores of alpha crystals and some are lead. This was a small bearing cast in sand.

Fig. 11—Same as Fig. 10, but magnified 200 diameters. The black spots are the soft cores, the rounded dark gray spots are lead and the bright spotted streaks are eutectoid, filling in between the alpha crystals.

Fig. 12—Bronze bearing, cast in sand, containing about 6 per cent tin, 1.5 per cent lead and 2 per cent zinc, etched with ammonia and

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hydrogen peroxide and magnified 20 diameters. This is fairly typical of the bronzes low in tin.

Fig. 13—Same as Fig. 12, but magnified 200 diameters. The soft cores are here broader than in Fig. 11 on account of the lower tin content, and there is hardly any eutectoid.

Fig. 14—Same alloy as Fig. 12, etched and magnified similarly, but cast in a chill, giving a finer grain.

Fig. 15—Same as Fig. 14, but magnified 200 diameters. This shows a much finer grain than Fig. 13, and compared with Fig. 7 the soft cores are seen to occupy more of the field of view, with less of the harder bright alpha constituent, on account of the lower tin content.

Fig. 16—Manganese bronze, etched with ferric chloride and magnified 20 diameters. The large grains of varying shade are beta solid solution, and the small bright crystals, alpha.

Fig. 17—Manganese bronze etched with ferric chloride and magnified 200 diameters. This is a rather hard strong bronze; softer manganese bronze would show more of the bright alpha constituent.

Fig. 18—Tensilite, etched with ammonium persulphate and magnified 100 diameters. This structure is similar to that of manganese bronze.

Fig. 19—Titanium-aluminum bronze, cast in a chill, etched with acid ferric chloride and magnified 100 diameters. The light grains are alpha solid solution, and the dark constituent, which is harder and higher in aluminum than alpha, is probably a eutectoid of gamma and alpha solution.

Fig. 20—Same as Fig. 19, but cast in sand and magnified 400 diameters, showing the normal structure of the dark-etching eutectoid between the alpha crystals.

Fig. 21—Titanium-aluminum bronze casting that cooled too slowly, magnified and etched like Fig. 20. On account of the coarse structure of the eutectoid, this alloy had a tensile strength of 59,000 lb. per sq. in. and an elongation of 10 per cent in 2 in., while the alloy represented by Fig. 20, of the same chemical composition, has a tensile strength of 70,200 lb. per sq. in. and an elongation of 20 per cent in 2 in.

Fig. 22—Same as Fig. 19, but cast in sand, giving a coarser grain.

Fig. 23—Structure of special aluminum bronze containing iron, etched with acid ferric chloride and magnified 20 diameters, showing alpha crystals arranged in network form as well as in small crystals. This alloy, with an elongation of 10.5 per cent in 2 in., had a tensile strength of 97,500 lb. per sq. in.

Fig. 24—Structure of alloy similar to that illustrated by Fig. 23, but containing also a little manganese. The network arrangement of the alpha crystals is not noticeable here, as in Fig. 23, and this alloy, with an elongation of 10.5 per cent in 2 in., had a tensile strength of 110,700 lb. per sq. in.

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FIG. 1—CHILLED FACE OF PHOSPHOR BRONZE GEAR
Magnification—200 Diameters

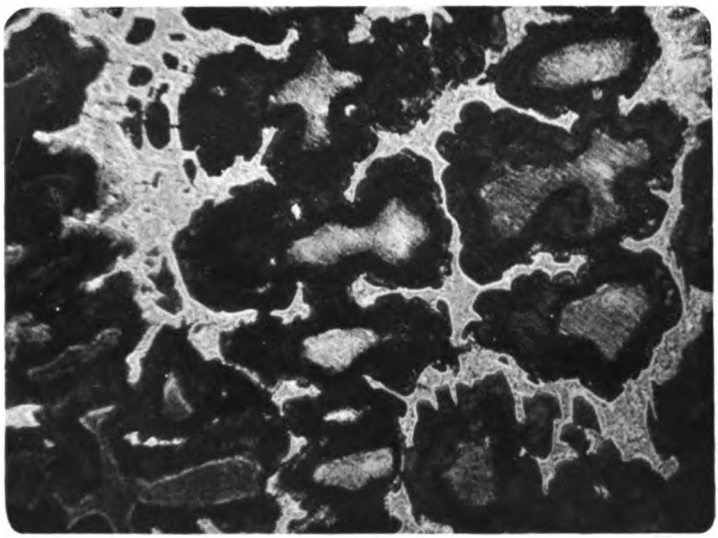

FIG. 2—SAME ALLOY AS IN FIG. 1 BUT CAST IN SAND
Magnification—200 Diameters

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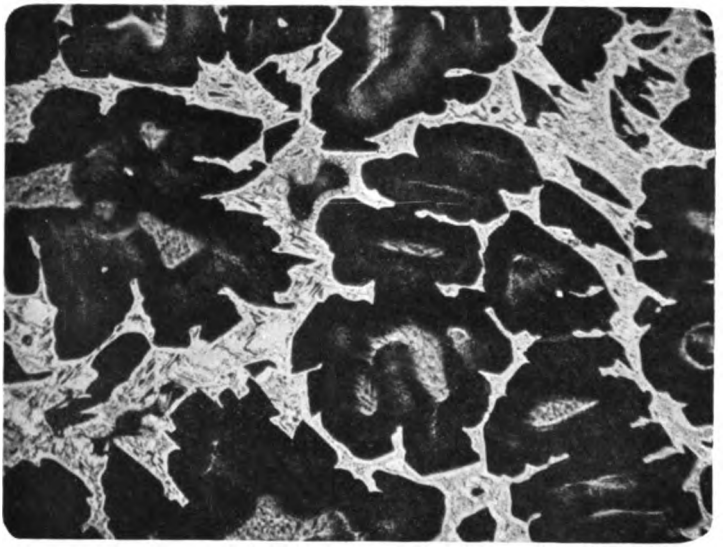

FIG. 3—BRONZE CONTAINING 13 PER CENT TIN AND 2 PER CENT ZINC
Magnification—200 Diameters

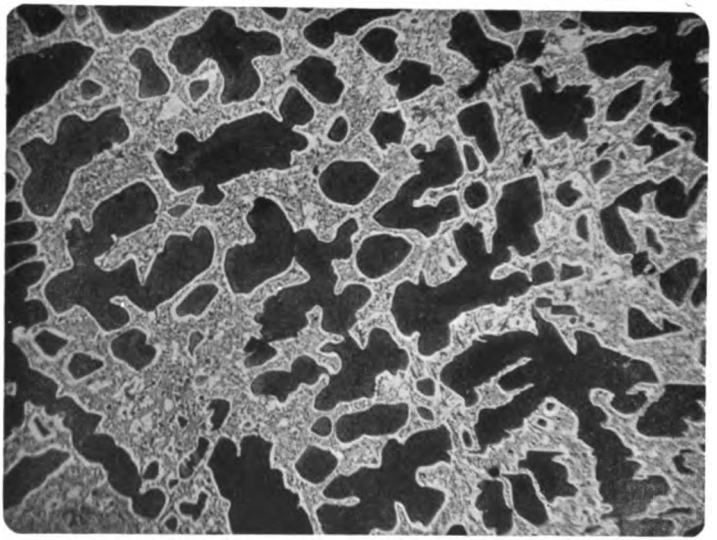

FIG. 4—HARD SPOT ON SURFACE OF PHOSPHOR BRONZE GEAR
Magnification—200 Diameters

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FIG. 5
HARD SPOT SHOWN IN FIG. 4
Magnification—20 Dia.
Reduced About 12 Per Cent in Reproduction

FIG. 6
GUN BRONZE CAST IN CHILL MOLD
Magnification—20 Dia.
Reduced About 12 Per Cent in Reproduction

FIG. 7
SAME ALLOY AS SHOWN IN FIG. 6
Magnification—200 Dia.
Reduced About 12 Per Cent in Reproduction

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FIG. 8
STRUCTURE OF ENGLISH WORM GEAR
Magnification—20 Dia.
Reduced About 12 Per Cent in Reproduction

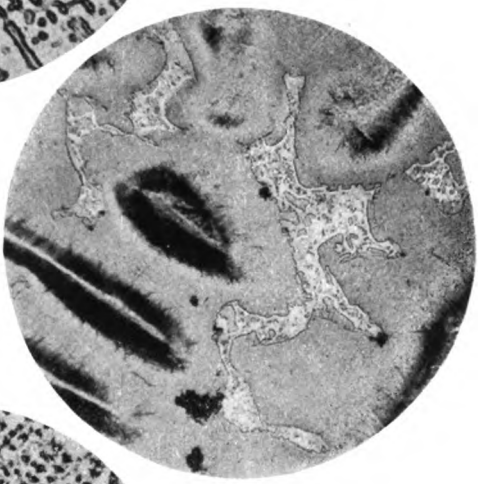

FIG. 9
SAME ALLOY SHOWN IN FIG. 8
Magnification—200 Dia.
Reduced About 12 Per Cent in Reproduction

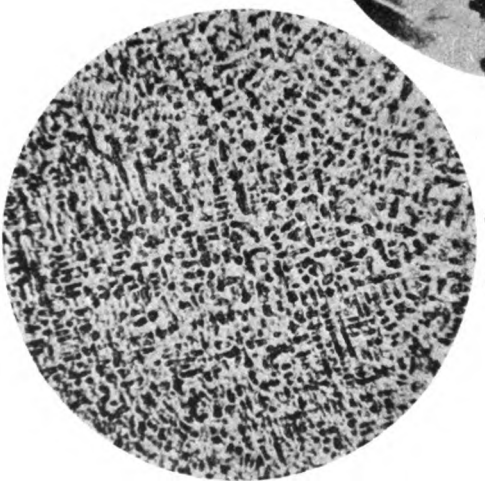

FIG. 10
STRUCTURE CLOSE TO SURFACE OF BEARING
Magnification—20 Dia.
Reduced About 12 Per Cent in Reproduction

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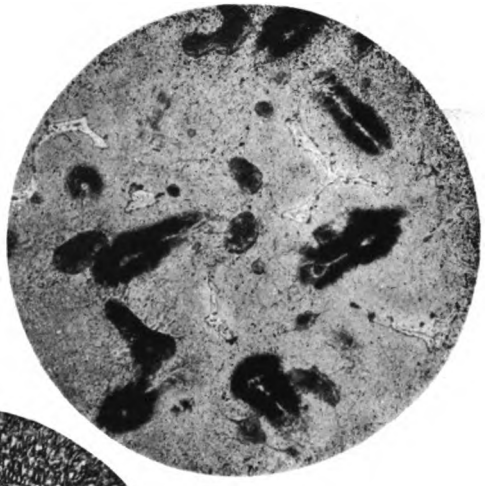

FIG. 11
SAME ALLOY SHOWN IN
FIG. 10
Magnification—200 Dia.
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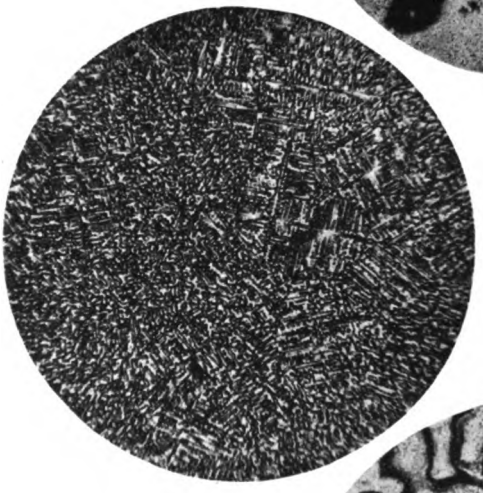

FIG. 12
BRONZE BEARING CAST IN
SAND
Magnification—20 Dia.
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FIG. 13
SAME ALLOY SHOWN IN
FIG. 12
Magnification—200 Dia.
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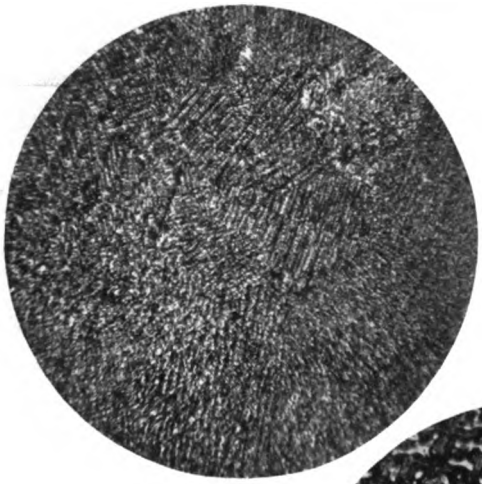

FIG. 14
BRONZE BEARING CAST
IN A CHILL
Magnification—20 Dia.
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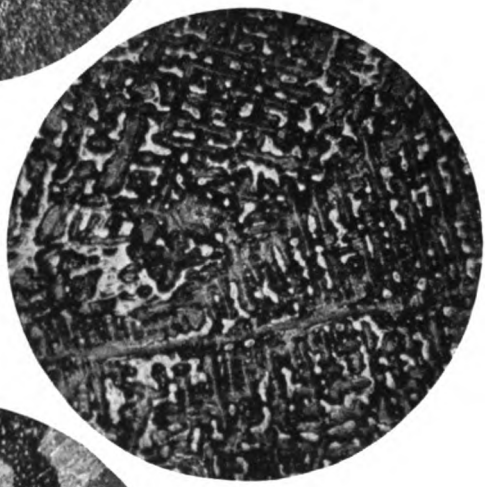

FIG. 15
SAME ALLOY SHOWN IN FIG. 14
Magnification—200 Dia.
Reduced About 12 Per Cent in Reproduction

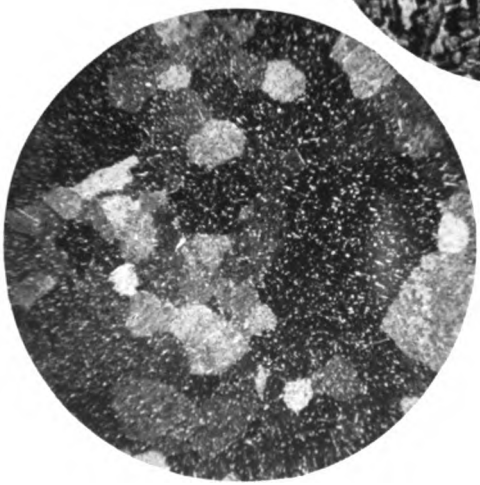

FIG. 16
MANGANESE BRONZE ALLOY
Magnification—20 Dia.
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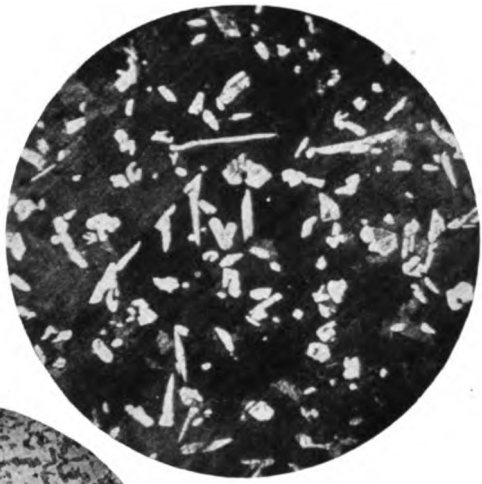

FIG. 17
MANGANESE BRONZE ALLOY
Magnification—200 Dia.
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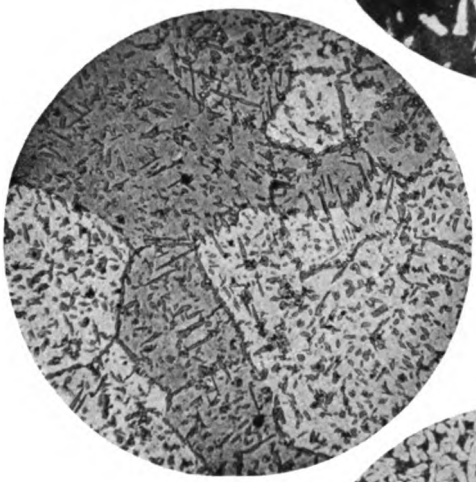

FIG. 18
TENSILITE ALLOY
Magnification—100 Dia.
Reduced 12 Per Cent in Reproduction

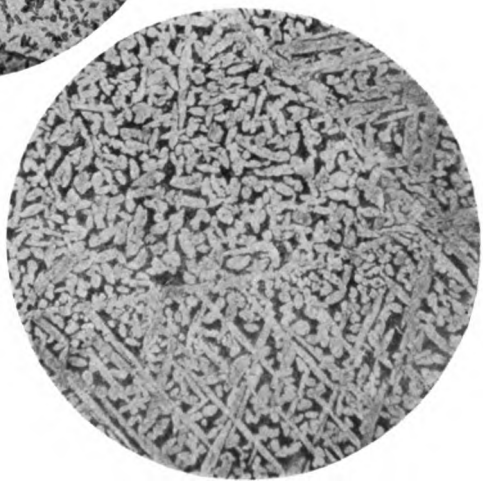

FIG. 19
TITANIUM-ALUMINUM BRONZE
CAST IN CHILL
Magnification—100 Dia.
Reduced About 12 Per Cent in Reproduction

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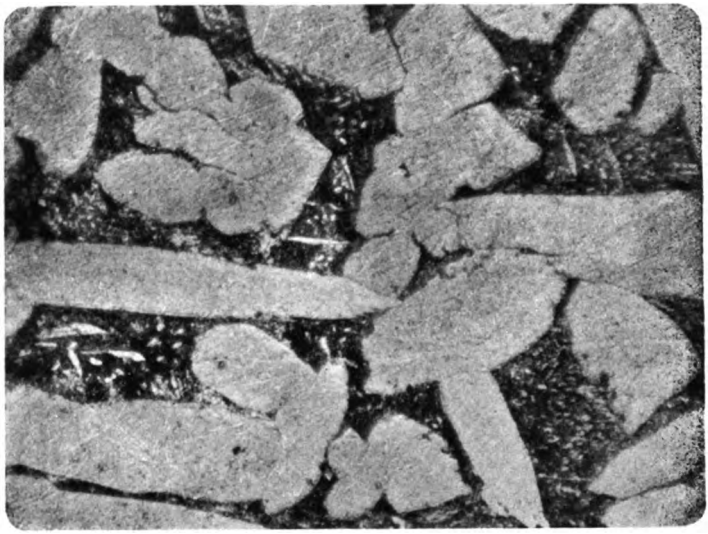

FIG. 20—TITANIUM-ALUMINUM BRONZE CAST IN SAND
Magnification—400 Diameters

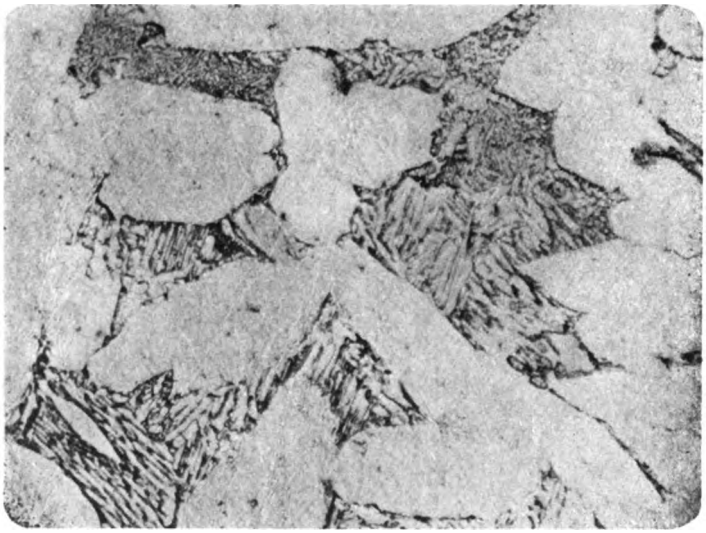

FIG. 21—SAME ALLOY SHOWN IN FIG. 20, THAT COOLED TOO SLOWLY
Magnification—400 Diameters

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FIG. 22
TITANIUM-ALUMINUM BRONZE, CAST IN SAND
Magnification—100 Dia.
Reduced About 12 Per Cent in Reproduction

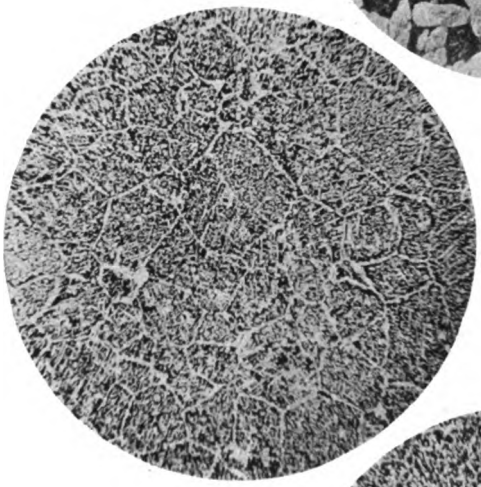

FIG. 23
ALUMINUM BRONZE CONTAINING IRON
Magnification—20 Dia.
Reduced About 12 Per Cent in Reproduction

FIG. 24
ALUMINUM BRONZE CONTAINING IRON AND MANGANESE
Magnification—20 Dia.
Reduced About 12 Per Cent in Reproduction

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DIFFERENTIAL SUBSTITUTES

By D. D. Ormsby

(Member of the Society)

Abstract

Substitutes for the conventional type of differential are considered under four classifications; namely, the free-wheel type, the crank and eccentric types, the spiral gear type, and the solid axle. Examples of each of these classifications are described and the advantages and disadvantages of some of the more practical ones discussed.

Considerable space is devoted to a discussion of the elimination of any form of differential whatever. Although such construction has advantages of eliminating the spinning of the wheels and assuring positive travel under all conditions, the author believes the disadvantages too great to be overcome.

The paper mentions some interesting experiments conducted by street railway engineers in connection with using differentials for street cars, to eliminate the corrugation of rails and wheels, as well as to economize in power consumption. It was indicated that with a street car equipped with a differentiating mechanism about one-half the power consumed by a car equipped with solid axles would be saved.

The author believes that the ultimate differential will be one that compensates freely for the difference in speeds of the rear wheels when the car diverges from a straight course, and is so constructed that it will be impossible for either wheel to spin when the other has lost traction.

OPERATION OF CONVENTIONAL GEARS

The year 1915 stands out more than any other as the one in which considerable effort was made to correct the inherent faults of the conventional type of differential. These efforts have extended into the present year and have been attended with much interest on the part of automobile engineers as well as gear manufacturers. The outcome of present experiments will be of interest to all.

Before discussing the substitutes that have been designed to rectify the faults of the spur and bevel type differential, let us consider whether the standard types fail through inefficiency or because of too great efficiency. I believe the latter to be the fact. With these conventional gears we find that in making a turn the outer wheel, which travels the greater distance, accelerates and the inner wheel retards, this being the desired action. There is another difference which the standard differential takes care of; that is, a difference in load which is caused by unequal traction of the wheels, by allowing the wheel having the lesser load to revolve faster than the one having the greater load.

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Therefore instead of being inefficient, the conventional differential is too efficient in that it differentiates for all differences, whereas the automobile engineer wants it to take care only of the unequal velocities of the rear wheels. It becomes inefficient as a means of transmitting the power to the rear wheels if the wheels have unequal traction. Many intended substitutes for the present types of differential have been invented to correct this defect of differentiating for unequal traction of the wheels. The substitutes can be divided into four classes: The free-wheel type (Figs. 1, 2 and 3); crank and eccentric (Fig. 4); spiral differential (Fig. 5); and the solid axle.

FREE-WHEEL TYPES

Fig. 1 shows the Hedgeland solid anti-skid automobile axle, of which little has been heard for some time. As the name indicates,

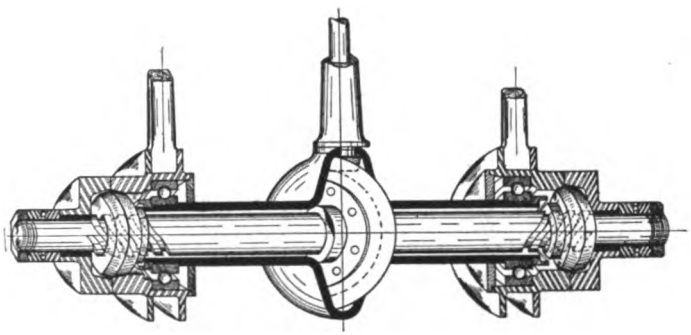

FIG. 1—HEDGELAND ANTI-SKID AXLE

this axle was constructed with a continuous or solid axle, having a thread cut on both ends. To these threaded ends were attached cone clutches having a double taper. When the power was applied to the axle these clutches were forced one way or the other by the thread on the shaft and gripped a cup that was attached to the wheel, thus transmitting the power from the shaft to the wheels. When a car equipped with this device was making a turn or diverging from a straight course, so that the outer wheel was obliged to travel faster than the inner wheel, the outer wheel was automatically declutched and ran freely on the axle around the clutch on the shaft. Fig. 1 shows the mechanism of this construction.

Fig. 2 represents a design similar to the one in Fig. 1, and the same principle is utilized, namely, having a loose wheel when the car is diverged from a straight line, so as to differentiate for the unequal velocity of the two rear wheels. It is practically the same as the Hedgeland design, with the exception of the clutching mecha-

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nism, in that it employs a solid axle, to the center of which is fastened the bevel drive gear and on the ends of which the clutches are applied. In this case the clutches consist of two cams on each side, keyed or secured by some other means to the shaft; when driving forward or backward they force rollers out against a drum to which is attached the sprocket or rear wheel. By wedging these rollers against the drum the power is transmitted from the cam to the drum and thence to the rear wheels. Both wheels are gripped alike when the car is being propelled in a straight line but when the car diverges from a straight line, as in turning a corner, the outer wheel runs faster than the inner wheel, forcing the rollers back so that it is released from the cam and hence from the shaft, and allowed to rotate faster than the inner wheel.

The substitute shown in Fig. 3 is of similar design to the one shown in Fig. 2, in that cams and rollers are employed, but it is mounted in the conventional position instead of in the hubs of the

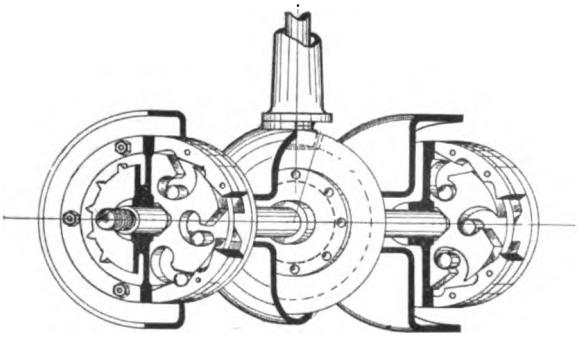

FIG. 2—FREE-WHEEL TYPE OF AXLE

wheels. Its action, however, is the same; when the car is being driven in a straight line both cams are clutched to the drum which drives both wheels at the same speed. When the car ceases to run in a straight line the outer wheel travels ahead of the inner wheel, declutching the cam attached to the shaft of the outer wheel.

Defects of Free-Wheel Differentiation

The action of the free-wheel type differential is such that it eliminates the inherent defect in the standard differential; that is, differentiating when the wheels have unequal traction. Yet in some ways its action is not entirely satisfactory. In the first place, with unequal size tires it drives more on the large tire, because the wheel with smaller tire has to rotate faster to make up for the distance traveled by the wheel having the larger tire. Secondly, in making a turn all the power is applied to the inner wheel and hence

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must be transmitted through one shaft. This throws an extra heavy load on the inner wheel, especially when making a turn on a steep hill.

According to the experience of a car manufacturer, another serious defect is the shock of applying the brake when the car is not running in a straight line. When making a turn the outer wheel, being free from the axle, is more sensitive to the brake than is the wheel that is transmitting power. The outer wheel will be retarded much more rapidly than the inner wheel. There is considerable shock when its speed becomes less than that of the inner wheel. The differentiating mechanism is obliged to absorb this shock.

CRANK AND ECCENTRIC TYPES

The second type of differential substitute is shown in Fig. 4. For convenience I will call it the crank and eccentric type. I have not been able to obtain the name of the inventor of this differential and

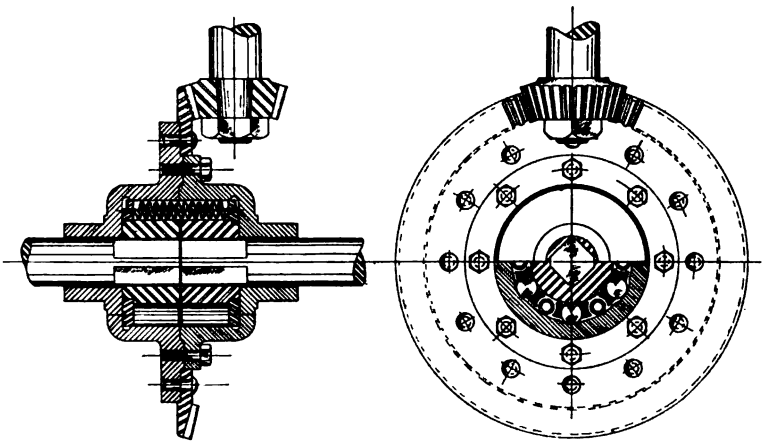

FIG. 3—CAM AND ROLLER TYPE OF FREE-WHEEL MECHANISM

believe it has not been tried out to any great extent. It is composed of two axle shafts, *C* and *D*. At the ends of these shafts are turned two eccentrics, to which are attached four cranks, *H*, *K*, *J* and *L*; *H*, *K* being attached to the eccentrics on shaft *D* and *J*, *L* to the eccentrics on shaft *C*. These cranks carry bosses which are bored and through which are passed sliding pins *E* and *I*, upon which the cranks are centered and oscillate. The cranks *L* and *H* are attached to sliding pin *E* and *J* and *K* to sliding pin *I*. These pins are allowed to slide in slots in the case *F*, which is made in two halves bolted together.

The action is as follows: The power applied to the bevel driving pinion *A* is transmitted to drive gear *B*, which is attached to and therefore rotates the case with the gear. When the car is traveling in a straight line, the sliding pins *E* and *I* rotate with the case, and

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the whole differential mechanism is rotated. When the car diverges from a straight line and the wheels have unequal velocities, the outer wheel speeding up oscillates the cranks by means of the eccentrics on the shaft, the cranks being allowed to oscillate around the center of the axis of the shafts and the eccentrics placed at such an angle in relation to each other that the required reverse direction of motion of the shafts is obtained. This allows the outer wheel to speed up and the inner wheel to become retarded, but because the angles of the eccentrics are so sharp, when one wheel loses traction it does not spin. I have never seen one of these differentials tried out and do not know how efficient it is, but it is ingenious and unique.

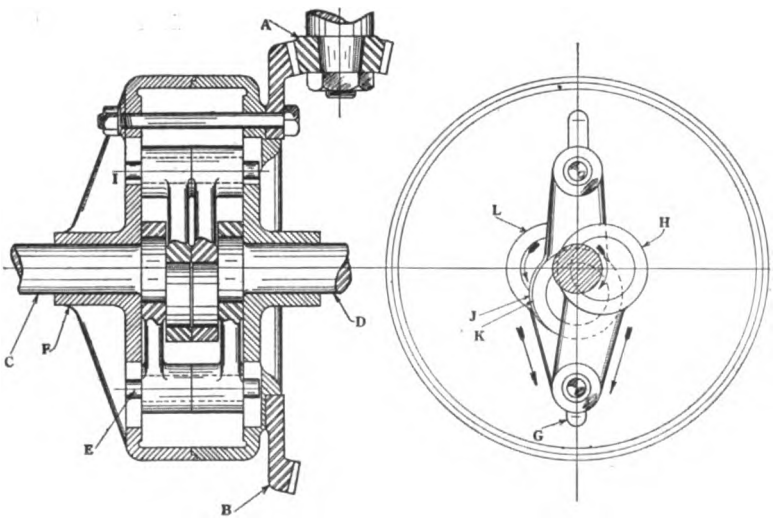

FIG. 4—CRANK AND ECCENTRIC TYPE OF DIFFERENTIAL

SPIRAL-GEAR DEVICE

The third type, which has received considerable attention, is the spiral gear, Fig. 5. This is similar to the conventional type, except that its differentiating gears are made up of a train of spiral instead of bevel or spur gears. Its action is the same in principle as that of the bevel or spur gear types, but the spiral gears are placed in such a manner as render the differential train inefficient, so that it is impossible to spin one wheel without rotating the other. Such a design should be the ultimate solution of the present differential problem, for it is evident that both wheels should be connected to each other at all times by some means and that the power should be applied to both wheels under all conditions, which is accomplished in this design.

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DISADVANTAGES OF SOLID AXLES

The fourth "substitute" eliminates the differential altogether. This construction obviates the spinning of the wheels, insures a positive drive of the wheels under all conditions and makes possible a reduction in cost. The disadvantages of this construction are, however, in my opinion, too great to be overcome.

The theory is advanced that the slippage is greater in making a short turn than in making a turn of larger radius through a given angle. With the solid axle slippage of the inner or outer, or of both wheels must always occur when the car is making a turn. It is hard to demonstrate exactly what happens. I have figured just how much slippage there is in making a turn of 90 deg. with a cer-

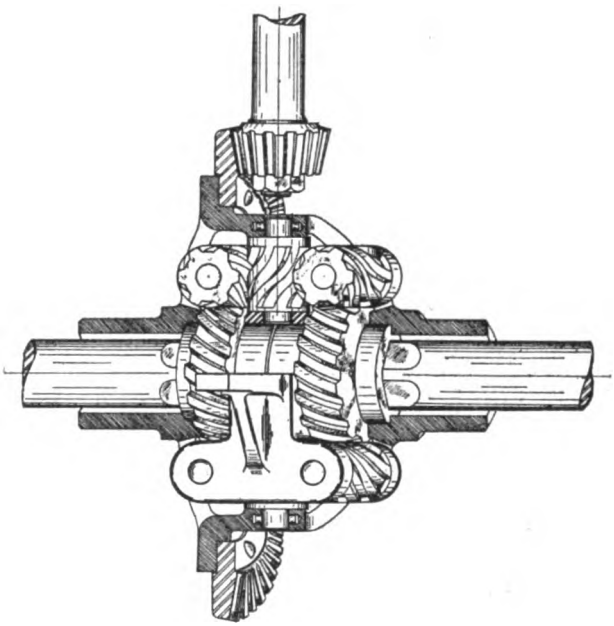

FIG. 5—DIFFERENTIAL WITH SPIRAL GEARS

tain light touring car and a well-known runabout. I found by measuring the amount the front wheels can be turned in steering, that the touring car can be turned in a radius of 17.13 ft. (205.6 in.). This is the radius at which the inner wheel turns. The standard gage being 56 in., the outer wheel would turn in a radius of 261.6 in. The distance traveled by the outer wheel in a 90-deg. turn is 410.9 in. The inner wheel would travel 322.9 in.; a difference of 88 in. to be compensated for by the slippage of the wheels. The total circumference of the touring car 32-in. tire is 100.5 in. The amount

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of slippage divided by the circumference of the tire gives the proportion of a revolution (0.87) that the wheel has to slip in making a turn of 90 deg.

The shortest radius at which the runabout examined can be turned is 13.9 ft. or 166.7 in. This is the radius at which the inner wheel turns. The outer-wheel turning-radius is 222.7 in. In making a turn of 90 deg. with this radius the outer wheel travels 350 in. The inner wheel travels 262 in. in making the same turn, the difference being 88 in., the same as with the touring car. I proved to my own satisfaction that the difference in travel between the inner and outer wheels in making a turn at a given angle depends upon the gage and not upon the radius of the turn. Taking the runabout tire diameter as 30 in., the circumfercence is 94.2 in. and the amount the wheel would have to slip 88 in. This divided by the circumference of the tire gives the part of the revolution that the wheel has to slip in making a turn of 90 deg., which is 0.93. No matter how great the radius, the amount of slippage through a given angle is always the same, for the difference in travel depends upon the gage and not upon the radius. Therefore with a solid axle no matter how slight the divergence from a straight line, an extra load, caused by the difference in travel of the two wheels, is always thrown upon the rear axle, wheels and tires. With a larger radius the slippage is not so noticeable, of course, as it occurs through a greater space.

Another objectionable feature in a solid axle is the additional power consumed in making a short turn, by being obliged to slip one or both wheels, for with a heavy loaded car or truck, when the slippage must take place in a short distance, the amount of slippage is greater in proportion to the distance traveled than it is on a long turn and the ratio of slippage is greater in proportion to the percentage of distance rolled by the wheels. Practical demonstration of this has been given by different manufacturers. One of these said: "A truck equipped with a solid axle could not be pushed by hand with the wheels cramped at an acute angle, whereas it could be pushed easily in a straight line. The truck was harder and harder to push as the wheels were turned, until a point was reached at which it could not be moved."

Another condition for which the solid axle does not compensate is unequal size of tires. Both tires on the rear wheels will not always be changed simultaneously. Suppose a truck is equipped on the rear wheels with two tires, one of which is 1 in. less in diameter than the other. Then with each revolution of the wheels one will have to travel 3.1416 inches farther than the other. The smaller wheel must slide this distance, or the larger wheel must slip part of the time to compensate for the lesser distance traveled by the smaller wheel. This certainly would cause excessive wear on a new tire.

DIFFERENTIAL IN STREET CARS

Street car manufacturers are considering the adoption of differentials for street cars, to eliminate the corrugation of rails and wheels,

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as well as to economize on the consumption of power. G. M. Cameron, master mechanic of the New York State Railways at Rochester, N. Y., gave the author the results of an experiment he conducted recently on two street cars, one having a solid axle of the standard type and the other being equipped with loose wheels.

The first car was equipped with a solid axle and traveled a continuous loop for a distance of 23,562 ft. or 4.46 miles. The average speed was 10.11 m.p.h. The average kilowatt-hours per car-mile were 1.452, and the average watt-hours per 10 miles 99.5. The second car was equipped with an axle having a loose wheel, to allow for any difference in speed of the two wheels. The total distance traveled was 27,720 ft. or 5.25 miles. The average speed of the car was 10.10 m.p.h. The average kilowatt-hours per car mile were 0.737, and the average watt-hours per 10 miles 50.5, or about one-half the value consumed by the car equipped with a solid axle.

It is evident that greater economy is obtainable in the operation of a car equipped with a differentiating means to compensate for the difference in speeds of the rear wheels, not only in power consumption but in wear-and-tear on the parts, for if more power is consumed with a solid axle, there are greater loads on the working parts which are obliged to transmit the power.

THE ULTIMATE DIFFERENTIAL

It is my belief that the ultimate differential will be one which is free to compensate for the differences in speeds of the rear wheels when the car diverges from a straight course, and so constructed that it will be impossible for the wheels to spin when one or the other has lost traction. I believe that the present type differential, constructed so that the differential train will not be quite so efficient, would accomplish the purpose. The present differential is too efficient as far as the differentiating mechanism is concerned. If one were designed in which this excess efficiency could be curtailed, the result would be what is required. The spiral gear type seems to fulfill the requirements more adequately than any of the others.

DISCUSSION

W. H. DIEFENDORF:—Mr. Ormsby has not mentioned one type (the New Process) of free-wheel differential, of which during the last year some fifteen thousand have been placed on the market. The majority of them have gone to car users and the only complaints made have related to the assembly of the differential in the axle. In the differential mentioned the inner wheel does the driving while turning the corner and the outer wheel runs ahead because of the traction between the tire and the pavement, of course throwing the entire load upon the driving wheel. So far, however, this does not seem to be serious, for the breaking of axles and shearing of keys have been eliminated in cars using these differentials.

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As I understand it the spiral-gear differential owes its positive-drive feature to friction between the parts, which must be expensive. The free-wheel type I have mentioned has very little friction. A car with the spiral-gear differential will have a strong tendency to straighten after taking a turn, showing there is friction in the differential. The bevel-gear type has this tendency to a lesser degree. The differential to which I refer hardly has this tendency at all. In fact, the steering-wheel can be set and circles cut with hands off the wheel. It is some little time before the wheels of the car begin to straighten out.

D. D. ORMSBY:—The spiral-gear differential we have been making has about the same amount of backlash when the car starts as has the ordinary bevel-gear type. I can give exact data in only one instance. I have driven one of these differentials in a large Kissel 1912 touring car about 12,000 miles. At the end of that period I did not notice any more backlash in the propeller-shaft or rear wheels than when I started. I think that could be explained by the fact that when the spiral differential locks it locks practically positively, so that there is not much sliding friction and the wear is not excessive.

A. LUDLOW CLAYDEN:—I am not by any means a whole-hearted supporter of the solid axle, but I object to the normal type of differential.

I was one of the first, perhaps, who rode in the first car equipped with the Hedgeland axle. The outstanding fault of the Hedgeland axle was that there was too much backlash, too much movement between going ahead and with the engine driving the car. When the throttle was closed and the car began to drive the engine there was a perceptible pause and then a rather severe bump as the wheels took up the drive. I think the backlash was something like half a revolution of the rear wheels. That was due to poor workmanship, but it also suggested a possible fundamental drawback. In the Hedgeland axle every time the clutches settled home they must have worn to some slight extent. That means that if the Hedgeland type was made so that there was practically no backlash when new, backlash would slowly develop.

I would like to ask Mr. Ormsby what backlash exists in the spiral type of differential when it is new? Does the inefficiency of that type of differential produce wear and does the backlash tend to increase? The spiral type of differential appears to be about the best mean between the too efficient differential and the too inefficient solid axle.

EXPERIENCE WITH SOLID AXLE

In 1914 I drove between 13,000 and 14,000 miles on a car with a solid axle. This car had a 90-in. wheelbase and a 48-in. tread; so it was about as severe a test as could possibly be devised for a solid axle. The car weighed between 1500 and 1600 lb. The maximum speed was about 50 m.p.h. and I have had the car up a little better

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than a 25-per cent grade. I compared that car carefully with others of similar dimensions with a differential, and I could not detect any perceptible difference in tire wear.

The greatest trouble was caused by a puncture in one rear wheel; it would almost surely tear out the valve. Although I had pretty good lugs to hold the tire to the rim, on several occasions considerable damage was done to the inner tube through my not realizing in time that one of the rear tires was practically deflated. Another drawback was that I could scarcely push the car with the wheels turned at an acute lock. That was rather a nuisance, because I kept the car in a garage that had an awkward entrance. It was essential always to have it in running condition to get it in and out, or get some one to help push it.

On the whole, after a year's experience, I think I would not use a solid axle again for a car of that size. But, on the other hand, that car could be run at high speed in fairly heavy mud without the slightest fear of a skid. It could be made to skid just like any other car, but you could go straight ahead with much more confidence than you can on cars of similar dimensions with the normal type of differential.

When Mr. Laycock of the Sheldon company suggested that it is possible to operate trucks without differentials, it was obvious that on account of its general dimensions the test of the truck was a little less severe than that of my small car. Furthermore, the possibility of one tire deflating was removed. Many parts of the automobile are theoretically as imperfect (and yet very satisfactory) as the solid axle. The general opinion of the engineers in this country is that the differential of the future is something just half way between the solid axle and the normal differential. I think that there is an extremely strong tendency to favor the spiral-type of gear.

ARTHUR M. LAYCOCK:—Mr. Ormsby mentions the slippage on the tires when rounding corners with the solid rear axle. He has worked this out carefully, but omitted to mention the enormous slippage of tires on a straightaway run with the conventional differential, over cobblestones and rough surfaces.

In going over rough road surfaces with a truck equipped with solid tires, the wheel is in the air for a considerable portion of the time, during which time it is naturally accelerating and coming down on the road at a different speed from that at which it left. The abrasive action on the tires under these conditions is distributed over a percentage of the total running time of the truck larger than when rounding corners with the conventional differential; in fact, P. M. Heldt in "The Gasoline Automobile" considers that the time taken in rounding corners on the average vehicle is a fraction of 1 per cent of its total running time. It would be much easier on tires to put in the blank differential and take care of the larger percentage of the running time of the truck than to introduce the complication of the pres-

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ent differential for only a small fraction of its operating time. In passing over bad road surfaces, there is absolutely no comparison between the two constructions. While most of the other positive forms of differential may be quite successful in light pleasure car and truck service, I have yet to see a positive form of differential operate successfully on a 3 or 5-ton truck, and this has been tried a good many times recently and with some of the best constructions. It is my firm belief that trucks of average wheelbase operating under the severest possible conditions will give much better service with the blank differential than with any other form of positive drive.

G. W. SMITH:—The relative percentage of slip permissible between outer and inner tires should be considered. It is my impression that rubber tires differentiate perfectly when the difference in distance traveled is small as compared with the total distance. It is apparent that a car can turn in a radius of sufficient length to result in a slippage of but 1 per cent of the distance covered; also that a car can turn in an extremely small circle that might result in as much as 5 to 10 per cent slippage. As nearly as I can determine the permissible amount of slippage without noticeable wear is between 2 and 4 per cent.

CHARACTERISTICS OF IDEAL DIFFERENTIAL

W. W. WELLS:—The ideal differential must have two characteristics that are so opposite in their nature that it seems to be impossible to combine them in one device.

First—When one drive-wheel turns faster than the other because the car is turning a corner or because the wheels are not of the same diameter, the differential must *divide the driving torque* between the wheels regardless of the *difference in speed.*

Second—When one wheel starts to turn faster than the other because it has lost traction, the differential must apply all the driving torque to the more slowly moving wheel, maintaining the *same speed* for both wheels regardless of the *difference in torque.*

How can a piece of mechanism distinguish between the two causes that produce the same result? How can it tell when the difference in speed is due to turning a corner and when it is due to the slipping of one wheel? A differential without any friction (hence 100 per cent efficient) satisfies the *first* condition and one with a large amount of friction satisfies the second. To illustrate this action, imagine a rear axle supported so that the wheels are free to turn and the driving gears are locked; then by turning the right-hand wheel *backward* the left-hand wheel is caused to turn forward. Power could thus be transmitted from the right-hand to the left-hand wheel through the differential and since in the bevel-gear differential this power is transmitted through two sets of gears and over one bearing of comparatively large diameter, the efficiency would be considerably less than 100 per cent. The torque acting on the left-hand gear of the differential would be perhaps 80 per cent of the torque applied by the right-hand gear.

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Any differential, whether of the spur, bevel, spiral or eccentric type, that absolutely prevents one wheel from spinning must have a zero or negative efficiency. Its action then is much like that of the free-wheel ratchet type in that it will not *drive* one wheel faster than the other, but it does allow one wheel to go faster if some outside force is the cause. It has the advantage, however, that it acts like a ratchet with an infinite number of teeth. Just as soon as the car starts on a straight course after making a turn, power is applied to both wheels again, whereas with a ratchet the pawl can be some distance from the bottom of its notch in the ratchet; the other wheel must then do all the driving until it slips enough to allow the pawl to engage.

ACTION OF VARIOUS TYPES

The inefficient type wastes some power on turns, must interfere somewhat with steering and necessitates designing each rear-axle drive-shaft to take the entire driving force. When driving two wheels of unequal diameter it delivers all the power to the larger wheel unless it slips enough to compensate for the difference in size. If the property of equalizing the torque is of comparatively little importance, the spiral-gear type of differential is perhaps the best, but it has two of the defects that Mr. Ormsby mentions as applying to the free-wheel axle.

The action of the eccentric type is nearly the same, but if it is made small enough to be interchangeable with the bevel-gear differential, the throw of the eccentric is much less than the diameter of the gear; hence, to deliver the same power the pressure on the eccentrics and pins will be proportionately greater than the tooth pressure and probably too great for the size of the parts that could be used.

The spur-gear type must be considerably less efficient than the bevel-gear type and has some advantage when one wheel has but little traction. It would also be possible to change the bevel-gear proportions so as to use a larger bearing and decrease the efficiency. In fact, I believe differentials could be built to divide the driving torque in almost any ratio desired.

If someone can invent a practical contrivance that will let one wheel turn about 33 per cent faster than the other, *but no more*, he will have something worth while; but that does not look like a promising field for invention.

Another substitute that looks feasible consists in attaching the wheels to a solid shaft by friction clutches or some other device that will slip when the torque reaches a definite amount. The capacity of each clutch should be something more than one-half the driving power of the engine on low gear. This would allow the full power of the engine to be utilized, but would never put the entire load on either half of the shaft. The action would be like a solid axle until the torque on one wheel became great enough to cause the clutch to slip; on a slippery pavement when turning the tires would slip; if the wheels had good traction the clutches would then slip.

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PNEUMATIC TIRE AND RIM SITUATION

By J. E. Hale

(Member of the Society)

Abstract

The author starts with the development of the pneumatic tire since its invention by Dunlop in 1888, and proceeds to show why several different types of tire construction are now in use. The merits of the three types of tires, namely, the clincher, straight-side and quick detachable, are discussed as regards energy consumption, traction, total mileage, cost-per-tire-mile, cushioning effect, reliability, ease of applying and service. The conclusions brought out show principally the advantages of the straight-side tire.

Statistics are offered to show the trend of the rim situation, and it is pointed out that it is just a question of time when the quick detachable clincher will cease to survive. It had a legitimate place during the development stage, but with the developed straight-side tires giving entire satisfaction, the author holds that there is no excuse for continuing the quick detachable clincher type.

In conclusion the paper reviews the rim situation both here and abroad and predicts that two-piece straight-side rims offer the next step forward in this department of the industry.

The pneumatic tire was invented by Dunlop in England in 1888. At that time rubber tires were used on carriages and foot-propelled velocipedes. As soon as the qualities of the pneumatic became appreciated, interest was stimulated to such an extent that Tillinghast invented the single-tube tire, the soft-bead clincher was brought out by Bartlett, and Dunlop himself devised the wired-on tire to fit the one-piece concave rim.

By 1898 the automobile industry had created a considerable demand for pneumatic tires. As this demand increased, it soon became apparent that the soft-base clincher tire was more practical than either the single-tube or the wired-on type. The single-tube tire had become popular in America, but as soon as the superior qualities of the clincher tire were appreciated it displaced the others rapidly.

CLINCHER-TIRE SHORTCOMINGS

The clincher tire had however certain disadvantages. It was hard to apply in large sizes. Considerable creeping trouble necessitated the use of several security bolts. Rim-cutting was prevalent on account of lack of skill in tire building and of poor fit on the rims that had not been standardized. Furthermore, the tires were ruined by being run in a deflated condition. These shortcomings fostered the introduction in America of the so-called straight-side quick-detachable

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rim, which was designed to take the wired-on tire in automobile sizes. The basic improvements claimed for these tires (which subsequently came to be called straight-side tires) were that they were easier to apply, security bolts were unnecessary, as the threaded valve with its rim nut and spreader was sufficient to prevent creeping, and they were less subject to rim-cutting.

By 1907 the detachable rim with a straight-side tire had become so much appreciated by the public that the clincher-tire manufacturers were obliged to furnish some sort of a detachable tire. The result was the quick-detachable clincher tire (called the Q. D. Clincher). The introduction of this tire was facilitated by the simultaneous introduction of the universal rim, which would take either a straight-side or a clincher tire by simply reversing the side rings.

While the quick-detachable clincher was a half-way compromise between the clincher and straight-side types, its introduction at that time was justified because the tire-building art was in a stage of development, not having progressed to a point where everybody was producing a successful straight-side tire. Up to the present time the straight-side, clincher and quick-detachable clincher tires have been exploited by their respective backers, and competition has stimulated development of each type. The basic structure of the pneumatic tire has not changed (except by the developments in cord tires) since 1907. The rim details have, however, been undergoing constant progressive development.

RIM DEVELOPMENT

In 1911 demountable rims began to receive commercial attention. In 1912 several manufacturers put them on cars as original equipment. The demountable rims were soon widely demanded by the public, which of course stimulated their further development. The early demountable rims were heavy. Competition has brought many new light-weight designs, some of which have been carried to such extremes that tire troubles result.

The preceding refers to American practice. In Europe the detachable rim and tire did not succeed in getting a foothold, so that as the industry expanded the use of the soft-bead clincher and one-piece clincher rims became firmly established. In Europe four circumstances saved the clincher from the competition of the detachable tire. First, rim-cutting, the worst trouble of the clincher tire, was mastered early. Second, the majority of car owners in Europe have chauffeurs who change the tires, which amounts to saying that a little saving in tire-applying effort does not interest them. Third, it is rather difficult, especially in England, to introduce new contrivances in competition with those firmly established. Fourth, the demountable wire wheel was one of the early developments in European automobile practice. Its use has been extended widely, so that if the quick tire-change idea had to be met, it was considered that the demountable wheel was satisfactory.

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NEED FOR EUROPEAN SIZES

In South America, Australia and South Africa European cars have predominated; naturally the use of millimeter-size (European) tires and rims is firmly established. However, during the recent period of trade activity by American manufacturers many inch-size clincher tires and rims have been introduced in these fields. This is most unfortunate because inch-size tires will be used on millimeter rims if the proper millimeter-size tires are not in stock, with certainty of rim trouble. The same thing is true of the application of inch-size tires on millimeter rims. This foreign business is expanding to such volume that the dimensional clincher-tire complication will, unless checked, ultimately react severely against American prospects. American car makers must discontinue the use of inch-clincher (except Ford sizes) and adopt either the millimeter clincher or inch-size straight-sides types.

The refinements accompanying the intensive development of the motor car have reached a point where even the weight of the light demountable rim is regarded with disfavor. A return to the simple light-weight detachable rim fitted directly to the felloe of the wheel seems likely. The car designers want to eliminate weight, particularly unsprung weight, at the periphery of the wheel, and to reduce the cost; the car owner desires a maximum of tire service with a minimum of tire trouble. Thus the situation boils down to a case of the survival of the fittest with the demountable rim, the demountable wheel and the simple rim mounted on a permanent wheel on the one side; and the straight-side tire, the clincher tire and the quick-detachable clincher tire on the other.

THREE TYPES OF TIRES COMPARED

The purpose of this paper is to point out the trend of the time in the tire world, and to venture a prediction concerning the future. In order to lay proper stress on the points to be brought out, it seems wise to discuss the merits of the three types of tires under the following headings: Energy consumption; traction; total mileage; cost per tire-mile; cushioning effect; reliability; ease of applying; and service.

Figs. 2, 4 and 8 (see appendix) show the straight-side, quick-detachable clincher and soft-bead clincher tires that will be compared. These are drawn to scale. The straight-side tire is shown on the wide standard rim. All three tires are identical in every particular above the line *H-K;* consequently the energy consumption and traction are the same for each type.

FACTORS AFFECTING TOTAL MILEAGE

The total mileage would be influenced by the structure of the tire below the line *H-K*. While the schemes of attachment to the rims have been largely perfected, the life of the tire is sometimes shortened

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by a form of trouble commonly called rim-cutting. In a well designed and properly built tire the chances of rim-cutting troubles from causes inherent in the tires themselves are remote. Imperfect rims and various forms of abuse are the actual causes. If a tire is punctured and ridden flat, the clincher casings and tubes will soon be ruined beyond repair. The clinch of the rim simply gouges into the carcass of the tire and cuts through the plies. At the same time, the tube gets under the toes of the bead, which mutilate it beyond repair. The structure of the straight-side tire, on the other hand, is such that the carcass is simply crushed against a comparatively wide rounding surface. Of course, the straight-side tire and tubes can easily be ruined by this form of abuse, but the chances are three to one in its favor, compared with the other type. The effect on the quick-detach-

TABLE I—TIRE EQUIPMENT (IN PER CENT) ON CARS PRODUCED 1913-1916

Type of Tire	1913	1914	1915	1916
Straight-Side Equip.	32	72	84	96
Quick-Detachable Clincher Equip.	40	28	16	4
Universal Rim	28	....	....	....
	100	100	100	100

able clincher is about half way between that on the clincher and straight-side tires. In this case the toes of the bead do not lift to catch the tube, but the clinch of the rim gouges into the fabric of the casing to some extent.

UNDER-INFLATION AND OVERLOADING

These are similar in their effects on the structure of the tire, being forms of abuse that aggravate any tendency to rim-cut. The effect is more pronounced on the clincher type than on the straight-side, because the flexing of plies is localized at the edge of the clinch with the former, while with the latter the flexing is distributed with less intensity over the gradual curvature of the rim.

TIRE-MILE COST OF OPERATION

The cost per tire mile is of course a function of the mileage and of the original cost of the tire. Inasmuch as the three types cost the same, the tire-mile cost is directly proportional to the mileage. Since the tires of the different types are alike above the line *H-K*, the service will average the same for all so far as the upper part of the tires is concerned. Any variation in the tire-mile cost will be dependent on the complications introduced by tire-base troubles.

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While no specific cost figures can be presented, there is no doubt that the tire-mile cost of the straight-side tire is the lowest. This is strongly emphasized by the domination of the straight-side over the quick-detachable clincher tires as shown in the percentages given in Table I.

CUSHIONING EFFECT

For a given tire section the cushioning effect is largely dependent on the relation between the load and inflation pressure. Lowering the inflation pressure gives the greater cushioning effect, but on the other hand fabric troubles are greatly increased by excessive flexing of the carcass. In practice it has been found that the best inflation pressure will permit the tread to be depressed 11 to 12 per cent of the tire section. Since all three types are alike above the line *H-K*, the cushioning qualities are the same if the tires are operated under similar conditions.

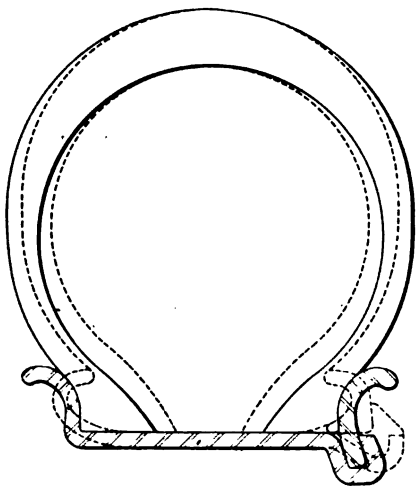

FIG. 1—STRAIGHT-SIDE TIRE (FULL LINES) AND QUICK-DETACHABLE CLINCHER TIRE (DOTTED LINES)

The difference in the rim-attachment structure of the two types has an effect on the cushioning properties. A straight-side and a quick-detachable clincher tire of the same size and designed to require the same quantity of raw materials, are superimposed in Fig. 1. This shows that the straight-side tire has the advantage due to the beads of the tire being more widely separated, resulting in a slightly larger diameter of tire section.

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SCOPE OF RELIABILITY

Reliability covers such features as freedom from instantaneous failures and safety considerations. Punctures, blowouts and fabric breaks occurring in the tread and side-wall portion of the carcass (above the line *H-K*) are on a par in all three types. Fabric failures at the point of contact with the rim are apt to be more in the nature of a concealed distress followed by sudden failure (blowout in effect) in the clincher and quick-detachable, while the corresponding troubles with the straight-side tires are easier to detect and provide against.

TABLE II—TIRE AND RIM EQUIPMENT ON CARS PRODUCED 1913-1916

Type	1913		1914		*1915		*1916	
	No. Cars	Per Cent	No. Cars	Per Cent	No. Cars	Per Cent	No. Cars	Per Cent
Q. D. Clin..	98,000	24	66,500	19	50,000	8	22,000	2
Str. Side...	79,500	19	172,000	35	266,000	40	529,000	46
Reg. Clin...	175,000	41	281,000	46	348,000	52	601,000	52
Univ. Rim..	67,000	16	None used	....	None used	....	None used	....
Totals...	419,500	100	519,500	100	664,000	100	1,152,000	100

*Figures are for manufacturers' season—July to July.

Blowing-off the rim is one of the troubles of bygone days. The early straight-side tires had a good many black marks against them on this account and clincher tires are still occasionally subject to this annoyance. This condition was one of the prime excuses for the introduction of the quick-detachable clincher tire. Because some of the early straight-side tires blew off the rim, certain car makers concluded that this type was not safe. It was not appreciated universally that the straight-side construction is a purely mechanical method of attachment and independent of inflation pressure. At the same time the detachable idea was so meritorious that all tire manufacturers were compelled to put out some sort of a "detachable" tire. Inasmuch as the soft-bead clincher had represented the best practice it was natural to combine the inextensible bead of the straight-side with the clincher idea in an attempt to provide a detachable tire that would not blow off the rim. In the undeveloped state of the tire-building art it was felt that in order to be perfectly safe a tire must hook under the clinch of a rim. The later development of the straight-side tire has however shown conclusively that the problem of safety is simply one of designing the tires to have proper strength at the

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weak point, so that at present, for every several hundred that fail by rim-cutting and blowouts, only one tire blows off the rim. How completely the straight-side tire has lived down original prejudices

TABLE III—SOFT-BEAD CLINCHER TIRES AND ONE-PIECE CLINCHER RIMS ON 1916 CARS

Name	Tire Sizes		Name	Tire Sizes	
Bell		3½	Monroe	3	3
Briscoe		3½	Metz		3½
Chevrolet	3	3½	Regal		3½
Dort		3½	Overland		4
Elco		3½	Pullman		4
Ford	3	3½	Scripps-Booth		3½
Maxwell		3½	Sphinx		3½

is evidenced by the fact that 529,000 new cars will be delivered on straight-side tires during the 1916 season.

EASE OF APPLICATION

The 3-in. clincher tires are no more difficult to remove and apply than the other types, but beginning with the 3½-in., each larger size is correspondingly more difficult to handle. In applying the quick-detachable clincher tires the bead needs considerable coaxing to get it properly seated under the rim clinch; and the carcass of the tire is often so stiff as to make it difficult to slip the locking ring into place. This is especially true with the oversize tires. The quick-detachable tire is frequently found to be frozen to the rim by rust, thus requiring much effort in removing it. This tire on a split rim is really a formidable proposition; application is difficult and removal almost impossible.

TIRE SERVICE

Service refers principally to the distribution of the sizes required by the trade. Owing to the extreme flexibility of the movements of the automobile, competition forces dealers to carry all the sizes in demand in clinchers, quick-detachable clinchers, and straight-side types.

Early in this paper it was stated that the tire and rim questions boil down to a question of the survival of the fittest. The statistics in Table II show very decidedly the trend.

It is just a question of time that the quick-detachable clincher will become eliminated. This tire had a legitimate place during the period of development, but with the straight-side tires giving entire satisfaction, there is no excuse for continuing the former type. Deal-

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ers have to carry the three kinds of tire with many sizes common to each type. With the quick-detachable clincher eliminated, nearly one-third the stocks could be discontinued, which would mean correspondingly better service on the other two types, less capital tied up in stocks, greater incentive to carry all sizes, and ultimately less expense to the whole industry, and more simplified production.

TABLE IV—STRAIGHT-SIDE TIRES ON 1916 CARS

Name	Tire Sizes				Name	Tire Sizes		
Abbott		4	4½		Jackson		4	4½
Allen	3½				Jeffery		4	4½
Alter	3½				Jones		4	
Apperson		4	4½		King		4	
Auburn		4			Kissel		4	4½
Briscoe	3½				Lexington		4	4½
Buick		4	4½	5	Lozier			4½
Cadillac			4½		Marion		4	
Case		4			Meteor			4½
Chalmers		4			Mitchell		4	
Chandler		4			Moline		4	4½
Chevrolet	3½				Moon		4	
Cole			4½		Oakland	3½	4	4½
Daniels			4½		Olds		4	
Davis		4	4½		Paige-Detroit		4	
Dodge	3½				Peerless			4½
Empire		4			Regal		4	
Enger		4			Reo		4	4½
Franklin		4	4½		Ross		4	
Glide		4			Saxon	3½		
Grant	3½				Studebaker		4	
Haynes		4	4½		Stutz			4½
Hollier		4			Scripps-Booth		4	
Hudson			4½		Sun		4	
Hupp		4			Overland		4	4½
Interstate		4			Velie		4	

TIRE EQUIPMENT USED BY MAKERS

To further explain the situation, I have prepared the Tables III, IV and V (from published information), showing by size and type the tire equipment used on the principal cars. Pneumatic-tired trucks are not included.

The total estimated production in 1916 is 601,000 cars using clincher tires and one-piece clincher rims. This estimate applies to the fourteen makers listed in Table III.

The total estimated production in 1916 is 529,000 cars using straight-side tires. This estimate applies to the 52 makers listed in Table IV.

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The total estimated production in 1916 is 22,000 cars using quick-detachable clincher tires. This estimate applies to the 17 makers listed in Table V.

From the three groups tabulated we observe (1) Clincher tires are confined to 3 and 3½-in. sizes;. (2) Straight-side tires are used in 3½, 4 and 4½-in. sizes. (3) The quick-detachable clincher tires are confined to the 4, 4½ and 5-in. sizes.

TABLE V—QUICK-DETACHABLE CLINCHER TIRES ON 1916 CARS

Car	Tire Sizes		Car	Tire Sizes		
Cunningham...	4½	5	Pathfinder.....		4½	
Fiat.....	4½	5	Pierce.....		4½	5
Locomobile....	4½	5	Premier.....		4½	
Marmon.....	4½		Stanley.....		4½	
McFarlan.....	4½		Stearns.....	4	4½	
Mercer.....	4½		Stutz.....			5
National.....	4½		White.....	4	4½	5
Packard.....	4½	5	Winton.....		4½	
			Westcott.....	4	4½	

CONCLUDING PREDICTIONS

The pneumatic tire has passed through its evolutionary stages. Future developments will be confined to refinements intended to eliminate the common tire troubles, and to increase the tire mileage.

The American standard inch clincher-rim contours, the British standard millimeter clincher-rim contours, and the American wide standard inch straight-side rim contours will undoubtedly survive all others.

Outside of the contours, however, rims are still in process of evolution. In the struggle to minimize weight, expense and tire troubles the existing demountable types may lose ground, particularly as the light-weight one-piece clincher and the two-piece straight-side rims, either on permanent or detachable wheels, apparently offer the next step forward.

EXTRA TYPES WILL BE ELIMINATED

We now have uneconomical duplication in sizes in three types of tires performing similar service, but the law of natural selection will effect the elimination of the unnecessary types. Probably clincher tires (in America) will be confined eventually to motorcycle and 3 and 3½-in. automobile tires. Straight-side tires will be used in some 3½-in. and in all larger sizes. The quick-detachable clincher will shortly disappear from use.

The export markets in absorbing American cars will assimilate American constructions more and more readily. Straight-side tires

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will be introduced into these markets during the coming season as original equipment on some American cars. This type will presently be appreciated as much abroad as it is in America. Thus we are to initiate a step that will result ultimately in a single world-wide standard for pneumatic tires and rims.

Figs. 2 to 11 are inserted for the benefit of persons desiring more detailed information on tires and rims.

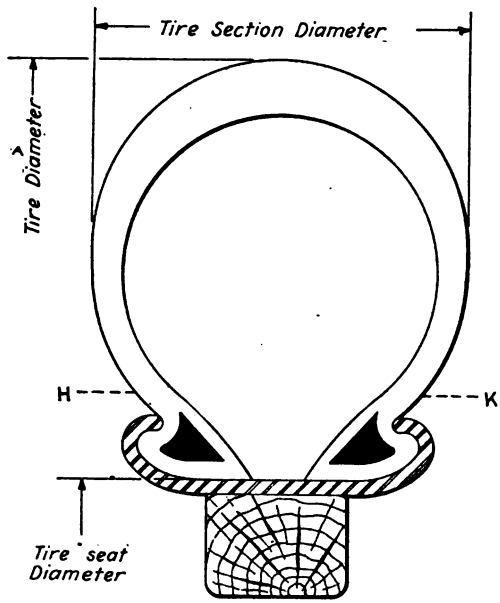

FIG. 2—SOFT-BEAD CLINCHER TIRE MOUNTED ON ONE-PIECE RIM

APPENDIX

The base portions of the tire in Fig. 2 have sufficient elasticity to permit their being stretched and pried over the edges of the rim when applying and removing it. The extensible bead core is made of semi-hard rubber.

The bead in Fig. 3 is understood to be the entire portion of the tire below the line *A-B*. The bead core is made of semi-hard rubber, which has just enough stretch to allow it to be pried over the rim. The tires are made about ½-in. smaller than the tire-seat diameter of the rim. This insures a snug fit against the rim and prevents the tube from pinching.

The internal bursting pressure of the air sets up stresses in the direction indicated by arrow *P*. These stresses are resisted at the

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hook of the clinch. In other words, a soft-bead clincher tire in its functioning is positively dependent on the air pressure to maintain its shape and position.

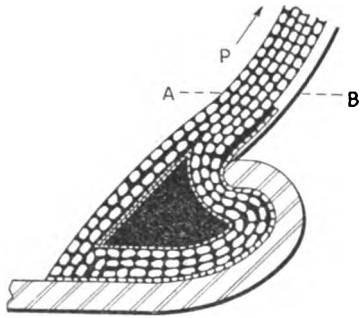

FIG. 3—DETAILS OF CLINCHER RIM AND BEAD

In Fig. 4 the side ring is cut in one place and can be removed by prying out with a screw driver. Figs. 5, 6 and 7 show enlarged

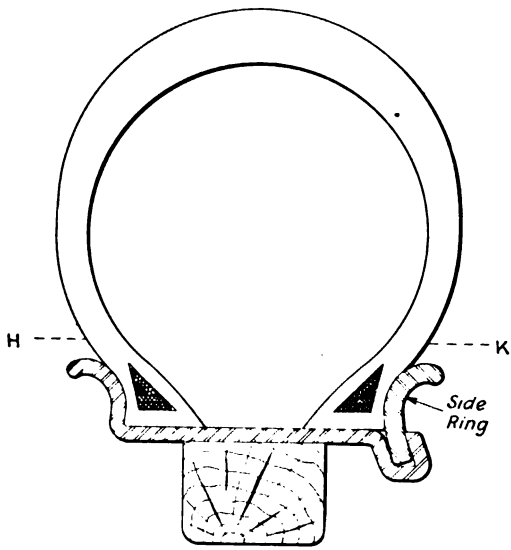

FIG. 4—STRAIGHT-SIDE TIRE MOUNTED ON TWO-PIECE DETACHABLE RIM

details of three prominent types of straight-side beads. Each bead is made inextensible by an endless reinforcement of braided wire (Fig. 5), twisted wire (Fig. 6), and wire cables (Fig. 7).

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In the type shown in Fig. 7 the fabric plies are tied under the inextensible reinforcing wires; thus the side wall stresses (P) are transmitted from the carcass fabric to these circumferential wires. It will be seen that a strictly mechanical method is used to fasten the tire to the wheel in contrast to the soft-bead clincher method, which is analagous to the action of an elastic band. The inside diameter of the tire is slightly greater (1/32 to 3/64 in.), than the

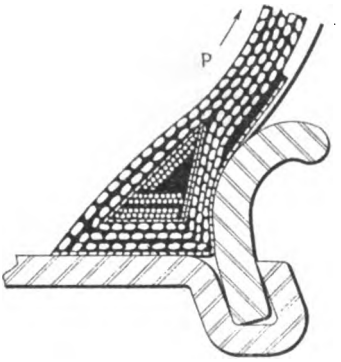

FIG. 5—INEXTENSIBLE BRAIDED-WIRE BEAD

corresponding rim diameter, thus permitting the tire to be applied and removed easily.

While the chance of pinching tubes under the beads of straight-side tires is remote flaps are used in the larger sizes, as a precaution.

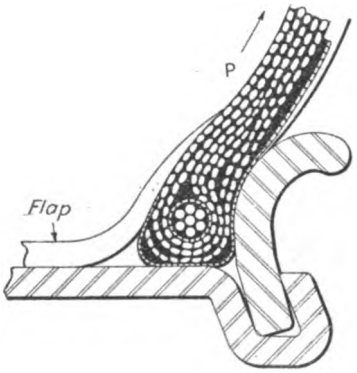

FIG. 6—INEXTENSIBLE TWISTED-WIRE BEAD

In the quick-detachable clincher rim, Fig. 8, the side ring is endless while the locking ring is cut in one place. In removing the base of the tire and the side ring embracing it must be crowded bodily

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toward the center of the rim far enough to clear the inner edge of the locking ring. The locking ring can then be removed by prying out with a screwdriver, after which the tire and the side ring can be removed.

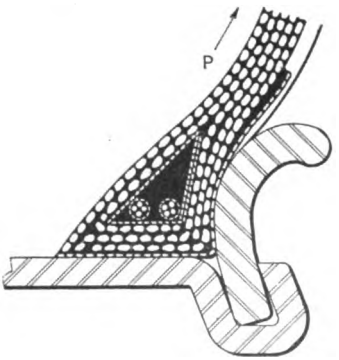

FIG. 7—INEXTENSIBLE WIRE-CABLE BEAD

Figs. 9, 10 and 11 show enlarged details of the bead structure of the quick-detachable clincher tire. By comparing Figs. 5 and 9 it will be appreciated readily that Fig. 9 simply shows a straight-side

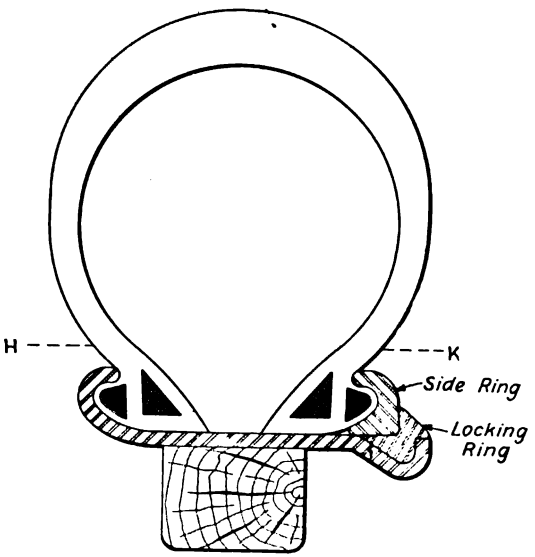

FIG. 8—QUICK-DETACHABLE CLINCHER TIRE AND RIM

tire with the addition of a clinch filler. This clinch filler is not tied into the body of the bead (by gripping with pliers the whole filler

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can easily be torn off the tire); consequently it is of absolutely no value in helping to hold the tire in the rim. Its real function is to fill the void that would otherwise exist.

Fig. 11 compared with Fig. 6 shows a similar situation, except that the outside ply of the carcass is laid so as to include the clinch

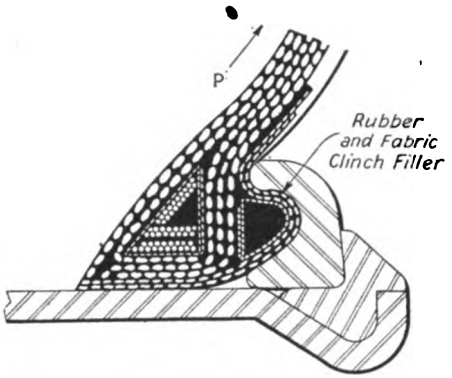

FIG. 9—INEXTENSIBLE BRAIDED-WIRE BEAD ON QUICK-DETACHABLE CLINCHER TIRE

filler. In Fig. 10 the quick-detachable construction closely resembles the regular soft-bead clincher construction. In this case, however, the bead core is made of a mass of cords impregnated with hard rubber

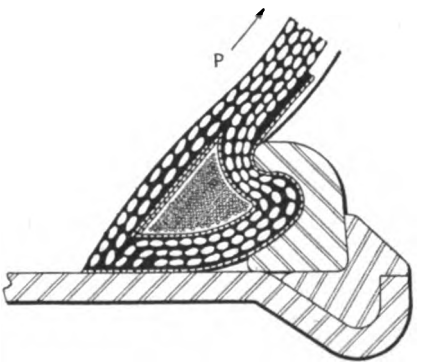

FIG. 10—BEAD OF PARALLEL CORDS ON QUICK DETACHABLE CLINCHER TIRE

and extending circumferentially around the tire. The bead of this tire is really semi-extensible.

DISCUSSION

A. L. CLAYDEN:—I believe that of all standardization the automobile industry of the world has before it the standardization of tires

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is the most important. It affects the largest amount of investment of any form of standardization not yet consummated.

The one fundamental difficulty is the two standards of measurement, the metric and the inch. At present I have no idea how this will be settled. Ultimately we shall find that the use of inches in some places and millimeters in others is really not a fundamental objection.

I believe that the demountable rim and the demountable wheel will both cease to exist. At present we carry either a spare wheel or a spare rim, both of which are heavy. We go through a mechanical process of fitting this spare, together with its inflated tire, to the axle, either the wheel on the hub or the rim on the wheel. That

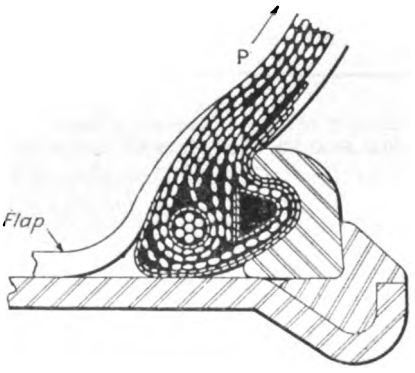

FIG. 11—HEAVY WIRE BEAD ON QUICK-DETACHABLE CLINCHER TIRE

means unscrewing and tightening two or three bolts and nuts at the least. We must carry about with us, in addition to the spare tire, a large amount of dead weight in the shape of spare metallic parts, either wheel or rim, as the case may be. I would like to see every automobile supplied with an air-tank maintained automatically at a proper pressure for tires. The same system as the Westinghouse, which is used on trolley and subway cars to maintain the brake pressure, could be provided. With any such arrangement a straight-side tire or tires of similar character can be changed just as quickly as a demountable rim or wheel, and the weight necessary to carry all the time will be less.

After studying the subject of demountable fittings, I am certain that we can consider the tire with a complete disregard of the rim situation. That is to say, if the straight-side tire is fundamentally a better type than the clincher it will, I believe, become universal. I think its adaptability to certain rims is not going to have any effect at all on the ultimate situation, say five years hence.

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LARGE SINGLE VERSUS DUAL SOLID TIRES FOR REAR TRUCK WHEELS

By W. H. Allen

(Member of the Society)

Abstract

This paper is mainly an argument in favor of the use of large, single rear wheel truck tires instead of smaller dual tires. Although the practice of using large singles is comparatively new, the author gives the results of experience and research to show the advantages of the newer method of rear tire equipment.

In developing his arguments in favor of single tires, the author goes into the history of dual tire application to show why it was necessary to use two tires in the earlier days of truck operation. As the necessity for increased carrying-capacity grew, tire manufacturers found the then existing single tire equipment inadequate, and they set about to develop suitable equipment to meet the new condition, the result being dual practice.

The method of attaching the earlier dual tires is shown to have been poor, resulting in circumferential creeping of the whole tire to a much greater extent as the width of the dual equipment increased. Inability to correct this weakness resulted in the conclusion that tires having such methods of attachment were not suitable when widths in excess of 4 or 5 in. were employed. The metal-base type of tire was developed to overcome the difficulty.

The author holds that dual tires are overrated, and believes that the practice of saying that dual equipment is capable of carrying loads double that of the single of which it is composed is open for discussion. The large single tire has its limitations and pending the results of further investigation, it seems advisable to consider 7-in. tires as the limit.

HISTORICAL

Research and practice, covering a somewhat extended period, have brought to the author the conviction that the use of large single tires, rather than a pair of small units on rear truck wheels, while still comparatively new, has nevertheless proved a progressive development. In order to discuss the subject thoroughly a brief historical review of the solid tire industry will be enlightening.

The use of dual or twin truck-tire equipment was inaugurated during the early development of the industry. It was offered as the then most practical way of meeting conditions, the exact severity of which was not thoroughly known. An accurate forecast of the variable operating conditions that followed the extension of the industry into all phases of commercial transportation was not then possible. Owing to such extension, the service has become more severe along lines of greater loads, higher speed and increased zone of

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activity to the point of overbalancing such bettered conditions as improved roadways, more skilful operators and improved suspension and design.

Early in the development of a solid tire that would satisfactorily meet the demands of commercial-truck service, three requisites were encountered. Listed in order of importance, these are: (1) Large carrying-capacity; (2) permanency of attachment; and (3) freedom from tendency to skid.

CARRYING-CAPACITY OF DUAL TIRES

The necessity for increased carrying-capacity soon appeared, and shortly tire manufacturers found no existing single-tire equipment adequate to meet practically and serviceably the new conditions. They at once started to develop suitable tire equipment, the result being that dual tires were recommended for all necessities above the range of single tire equipment. Later the practice was extended to rear wheels generally, because the carrying-capacity of tires applied in dual form was in some manner calculated to be from two and one-half to three times that of one of the units of which they were composed. Just why this was so considered has never been satisfactorily explained. It must therefore be assumed that the original load capacities, which are in effect to-day with no material change, were reasonably accurate. Practice and observation have confirmed the fairly general reliability of these schedules. Intelligible service or performance data are scarce, however, and so we find existing capacity schedules considerably deviated from in a number of instances. All these practices should be harmonized and a new and correct schedule should be established and followed.

EARLY METHODS OF ATTACHMENT

The early type of tire was attached by circumferential wires to provide substantially a compression of the rubber tire, in turn bringing its base into direct and firm contact with the wheel rim. The friction between the wheel rim and tire base thus obtained was sufficient to exceed the driving torque. Most prominent of these types was that in which metal cross-wires were embedded in the tire base at frequent intervals and extended laterally across the tire in such a manner that the ends were exposed. Circumferential wires were fitted over these ends under sufficient tension to draw the cross wires radially inward, thereby compressing the rubber underneath so as to secure and maintain a frictional contact between the tire and rim.

Then, as at present, the application was one depending upon friction to perform properly its intended function. As this type of tire was increased in width it was found impossible to secure the same degree of friction over its entire base, owing to the upward

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spring that took place in the center of the longer cross-bars, thus relieving the compression of the rubber under these bars. As a consequence the stability of attachment was reduced, resulting in circumferential creeping of the whole tire to a much greater extent than was true of those of narrower design. This movement resulted in rapid damage to the base and the tire was rendered useless prematurely, while its tread showed almost no wear in many cases. Inability to correct this weakness resulted in the conclusion that the design was not suitable for tires of greater than 4 or 5-in. width.

It was at this point that dual tires made their entry. Even triple applications were made in some cases. While results were somewhat more satisfactory, much room was still left for improvement. No similarly fastened tire has ever been free from the weaknesses explained, particularly when used as equipment for driving

TABLE I—CARRYING-CAPACITIES OF SINGLE AND DUAL TIRES

Size of Tire, In.	Type	Load* Applied, Lb.	Contact Area, Sq. In.	Distributed Load, Lb. per Sq. In.
36x3	Dual	2500	16.5	151.5
36x5	Single	2500	17.2	145.4
36x3½	Dual	3500	23.6	148.3
36x6	Single	3500	24.0	145.8
36x4	Dual	5000	26.9	185.8
36x7	Single	5000	31.5	158.7

*Corresponds to present schedule ratings for dual tires in sizes being compared.

wheels and when subject to varying and indifferent methods of application.

The third condition, namely that of tendency to skid, may be passed over with little discussion since this was wrongly considered as being wholly attributable to the design or character of tire. It is now generally recognized as being directly traceable to conditions of driving and braking.

NEW TYPE OF ATTACHMENT

Early in 1909 a new tire designed to correct the weaknesses of previous types was placed on the market. In general this type resembled similar tires used to some extent in continental Europe, although the latter were not advanced beyond the experimental stage. The American tire, commonly referred to as the "metal-base" type, embodied some entirely new features of shape and construction, and stands to-day, with minor refinements, as representative of the most advanced and successful practice. With the development of this tire, together with efficient standardization and accuracy in wheels, we

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have a condition whereby correct application is practically assured in every case. The frictional fit is adequate under the most severe conditions to resist any tendency for movement of the tire in any direction, regardless of size, since adhesion of the metal base to the felloe band increases directly with the tire width. This fact accounts largely for the perfection of attachment of the rubber tread to a steel rim or metal tire base, unattainable in any other known manner. The result is uniform and successful performance, as is amply evidenced by the results obtained. Weaknesses inherent in single tires of large size were overcome in this later type, so that such tires were at once brought into the field of practicability.

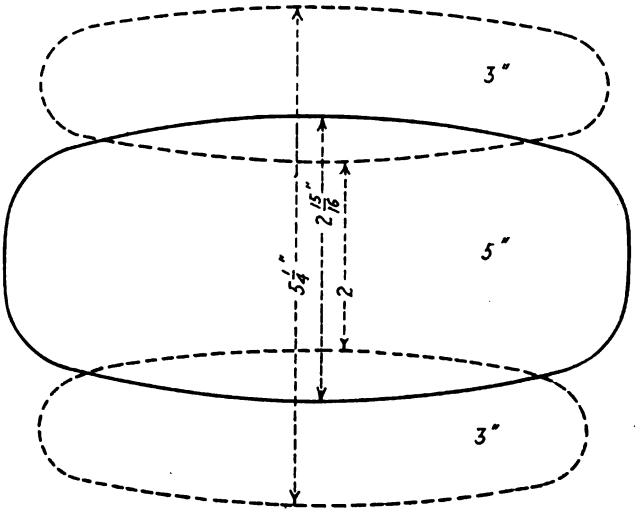

FIG. 1—RELATIVE BEARING CONTACT (2500 LB. LOAD) FOR 3-IN. DUAL AND 5-IN. SINGLE TIRES (HALF SIZE)

DUAL TIRES OVERRATED

The practice of rating two tires when applied dually at from 25 to 50 per cent more than the sum of capacities of the singles that make up the dual unit, is believed open to question, if not to direct criticism. No rule of theory or practice exists that will justify such practice. It seems manifest, however, that dual tires are somewhat overrated, and singles somewhat underrated, according to existing schedules. Such a schedule should in no case rate duals at more than the sum of the capacities of the single units employed.

SINGLE TIRES TO REPLACE DUALS

It is proposed that single tires of certain sizes replace dual tires of stated sizes. A comparison of carrying-capacity according to

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existing capacity schedules for the given sizes indicates that the proposed single tires are not equal in capacity to the dual tires they are expected to replace. In order to explain this point Table I and Figs. 1, 2 and 3 have been prepared. In each case the data are the composite results of several experiments made under ideal and similar conditions. Table I shows that in every case with equal loading the contact area of single tires exceeds that of the dual tires they are expected to replace, and that the load per square inch distributed over the contact area is reduced correspondingly with the increase in such area. Obviously, this is in favor of the single equipment. Figs.

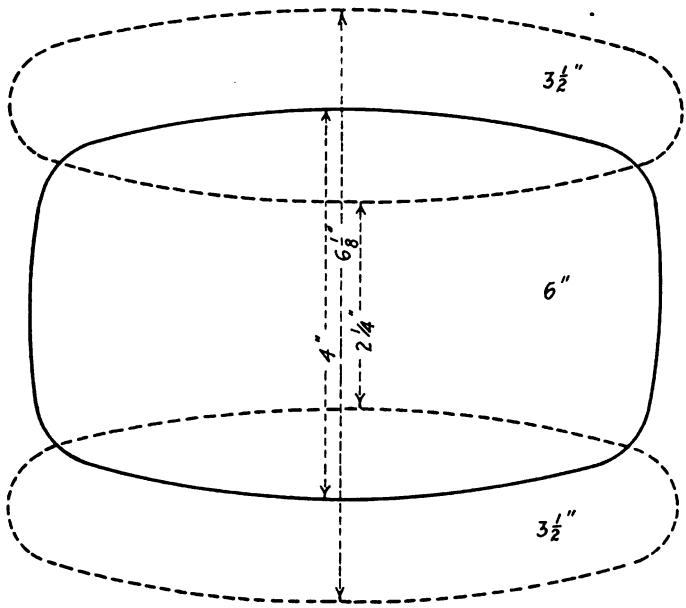

FIG. 2—RELATIVE BEARING CONTACT (3500 LB. LOAD) FOR 3½-IN. DUAL AND 6-IN. SINGLE TIRES (HALF SIZE)

1, 2 and 3 show to scale the area of contact. The reduction in width of tread ranges from 30 to 40 per cent.

EFFECTS OF OVERLOAD

All materials have a well defined limit of capacity for distortion in varying directions. Rubber is no exception, but its capacities are truly remarkable. In a popular sense its most natural enemy in solid-tire service can be considered as overloading. The more definite term is limit of displacement, which means that limit beyond which displacement or distortion causes breakdown of the fibrous structure

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to a point from which it cannot recover. This condition is often plainly evidenced by a torn or split appearance, but it is also often unnoticeable to the casual observer. Upon close inspection, however, a condition can be seen that is the forerunner of ultimate breakdown, this being indicated by permanent depressions and slight surface checking. Rubber in such a state has lost much of its ability to withstand further abuse, as it cannot absorb service shocks without more

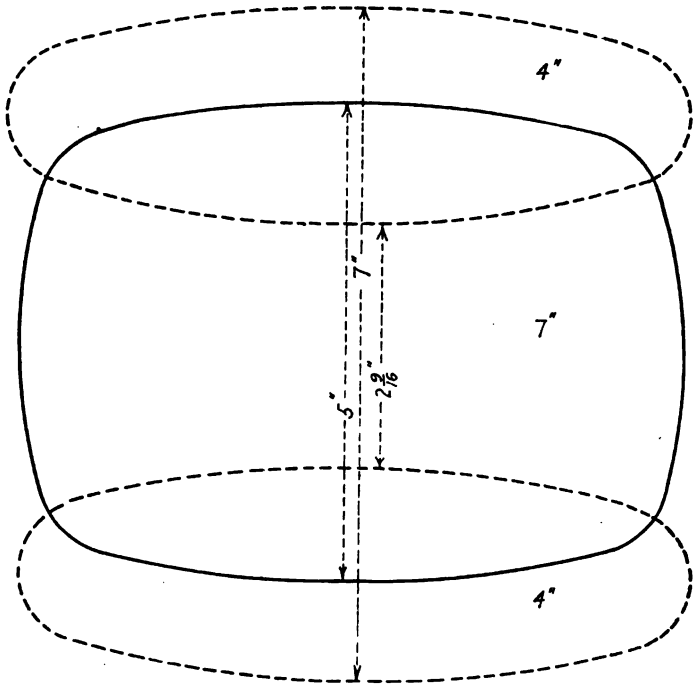

FIG. 3—RELATIVE BEARING CONTACT (5000 LB. LOAD) FOR 4-IN. DUAL AND 7-IN. SINGLE TIRES (HALF SIZE)

serious damage. It is in the same condition relatively as a metal part that has become fatigued through imposition of stresses beyond its capacity to absorb indefinitely.

SMALL DUALS INEFFICIENT

Dual tires composed of small units do not give satisfactory service for the reason that neither unit is of itself sturdy enough to resist momentary imposition of the total wheel load, such as occurs for instance when traveling over rough road surfaces, excessively crowned or furrowed roads. A condition is readily conceivable in such cases, wherein one of the small units is carrying the entire wheel load during a large part of the time, the load being shifted back and forth from one unit to the other. In view of the fact that

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such conditions do exist, it remains to alter the tire design or equipment in such a manner as will best take care of the case. This can be accomplished by the adoption of large single units, thus concentrating the load upon a tire sufficiently sturdy to absorb reasonable load inequalities. In the case of dual tires it is impossible to maintain an even load distribution on each small unit; the total load repeatedly and continuously being thrown from one to the other quickly results in rupture or other failure of either one or both of the units.

ADVANTAGES OF SINGLE TIRES

Numerous advantages are gained through the substitution of 5, 6 and 7-in. single tires for 3, 3½ and 4-in. dual tires. Chief among these are:

(1) Saving in tire cost ranging between 8 and 15 per cent.

(2) Saving in wheel cost. The reduced wheel cost is attributable to the narrower felloe stock and wheel rim required, together with whatever other saving may be made in connection with wheel design. The same reasoning also applies to metal wheels.

(3) Saving in cost of handling and applying one tire instead of two.

(4) Saving in wheel, tire and rim weight, resulting in possible greater operating economy and less unsprung weight.

(5) Larger tire that will adapt itself better to uneven road surfaces, compensate for excessive road crown, carrying its burden as a unit rather than alternately on one narrow tread and then on the other.

(6) Greater ease of fitting non-skid chains.

(7) Better tracking with front wheels (an advantage when tracks must be broken).

(8) Usually greater height of rubber tread, thus providing greater cushioning properties with consequently increased tire and vehicle life.

(9) Decreased strain on axle and wheel bearings as a result of decreased leverage obtained by the narrower wheel tread.

(10) Operating economies that will follow the use of large singles, if a deeper section than that commonly used be employed. The tire service in each case under observation has shown very material improvement.

LIMITATIONS OF SINGLE TIRES

The question will quite naturally arise, "If single tires in the sizes quoted possess advantages over dual tires composed of small units, why does not the same reasoning apply to single tires of sufficient width to replace the largest of dual tires?" Such practice may be possible, although developments to date have not been sufficiently conclusive in this respect to warrant the extension of the recommendation made beyond the ranges given. Perhaps it will be shown that a well defined limit exists, beyond which additional bulk cannot be successfully employed, from either a production or a service standpoint, in a solid sin-

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gle tire. Pending results of further investigation and of experiments being conducted, it seems advisable to consider 7-in. tires as the limit of practicable single equipment.

CONCLUSION

It is hoped that this paper will serve its definite purpose of converting truck builders and users to the advantages of adequate single tires over inadequate dual tires. Only ultimate good can then result to the future success of the whole industry. No truck manufacturer can dispute the fact that unserviceable accessories reflect to the detriment of his complete product. At best it can only be argued that the accessory manufacturer must stand back of his particular part; that adequate replacement of renewal burdens must fall upon him in case of failure. But what attitude does the user assume when his expectations of efficiency of a unit are not fulfilled in every essential? He criticizes not only the accessory that fails, but also the producer who allows inefficient accessories to become a part of his complete product. Continuous efficient operation is the condition desired and expected; adjustment service is only an unsatisfactory relief. After all, the user creates the demand and it is imperative that all of us see that he gets results. Such a course will favor expansion of the industry automatically and rapidly.

DISCUSSION

A. HARGRAVES:—The advantage derived by the use of large single solid tires instead of the small duals, as set forth in Mr. Allen's paper, should meet with the approval of the solid-tire manufacturer and purchaser interested in tire development and commercial efficiency. I concur heartily with the arguments made by Mr. Allen for the use of single solid tires, but believe that the practical limitation of this arrangement should not be confined to the 7-in. single size, but apply even more specifically to the larger sizes, such as 8, 10 and 12 in. Single solid tires of 8, 10 and 12-in. cross-section have demonstrated their practicability from both manufacturing and service standpoints. They have given exceptional mileage under extremely heavy duty and consequent overload. As with the smaller single tires, on excessively crowned or uneven roads the shock and load are distributed better over the large single steel base than in the case of the local application of stresses on a smaller dual tire.

Furthermore, these large tires afford greater traction surface and increased resiliency, because of the larger volume of rubber. Recent design of trucks to secure greater and greater capacity has created a demand for suitable tires to meet the greater stresses imposed and eliminate the inconsistencies of the dual arrangement.

Another feature incorporated recently in the large tire design, which can be considered an important factor, is a longitudinal grooving of the tread, giving the effect of two or more smaller tires on a single base and acting as a relief for the opposed stresses within

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the tire, thus preventing excessive lateral distortion at the point of road contact.

It has been demonstrated clearly in several cities that gasoline consumption has been reduced materially by the use of 8, 10 and 12-in. single tires in place of corresponding duals.

TIRES ON ARMY TRUCKS

W. H. ALLEN:—I have received some late information from the Mexican front regarding the use of single tires instead of dual. Probably trucks have never been operated continuously under more adverse road conditions. From Columbus, N. M., to Gibson's Ranch, a distance of 14 miles, the road is very sandy and full of holes. From Gibson's Ranch to Casas Grandes, a distance of about 120 miles, the road is substantially a desert trail, extremely rutty, full of stones and rocks and here and there cactus, mesquite roots and soapweed stumps. Below Casas Grandes, the roads are well-nigh impassable, the ruts are aggravated and the trails through the hills are practically rock and lava formation. One might think that constant pounding of the numerous truck trains over the same route would improve the road surface by crushing the irregularities. However, any dust that accumulates is, on account of the high winds peculiar to desert country, quickly blown off, leaving the rough rocky surface exposed.

On these roads ten truck companies of twenty-seven trucks each were in service. Five of these companies consisted of trucks equipped with 5- or 6-in. single tires all around. The other five companies consisted of trucks of the same capacity equipped with 3½-in. single front and 3½-in. dual rear tires. Inspection was made of tires on four trains that had equal mileage records. Two consisted of trucks equipped with singles, the others with duals. Not a tire on the trucks carrying single equipment needed renewal. All showed the severe service to which they had been subjected, but there was no sloughing on the sides and the wear was even. On the fifty-six trucks in the two companies equipped with dual tires, twenty to thirty tires were unsafe for another trip to and from the Front.

Practically all of the inside units of these dual tires were in serviceable condition; 99 per cent of the condemned tires were outside units. The reason for this is obvious. As one train consisting of single-tired trucks was followed by another with dual tires, some of the single tires ran in a rut from 3 to 5 in. wide: when the dual tires followed the inside tire ran unsupported over a rut while the outside tire carried the load and ran on the rough stones thrown out of this rut.

Truck captains soon realized the advantages of large single tires; one captain changed his tires around in the train so that all the trail-making would not be imposed on the leaders. He pointed out that the first three trucks in his train ran 265 miles fully loaded in the first, second and third positions in line, until he realized the conditions. The outside tires on all then had to be replaced after traveling only 1500 miles.

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REFINEMENTS IN TRUCK DESIGN

By H. D. Church
(Member of the Society)

Abstract

The author describes a number of detailed developments that took place during the working out of a line of worm-driven trucks.

The details of front axle and steering parts are dealt with at length, the reasons for the final constructions being clearly explained and the constructions themselves well illustrated.

Details concerning difficulty with the Hotchkiss type of drive on heavy trucks, troubles with drive-shafts and lubrication of the worm wheel are all covered thoroughly; spring-shackle construction and lubrication, radiator and hood mounting come in for detailed attention and the question of governors is interestingly covered.

Brief reference is made to the influence of unsprung weight, the differences between truck and pleasure car practice in this respect being pointed out.

In developing the present line of Packard worm-driven trucks, nine different experimental trucks were built and tested. The tests were run over a route in Detroit which was selected to obtain about 75 per cent of the distance over very rough, broken down block pavements. The trucks tested were overloaded continuously, were overspeeded during a portion of their testing, and were, as far as possible, kept in motion 22 hours a day, 7 days a week.

Under these severe conditions, any faults in design were rapidly brought to light, and this paper will deal briefly with some results obtained from such test service, and also with conclusions drawn from the service result standpoint of eight years of truck manufacture.

FRONT AXLES

In our worm drive development work we started with a front axle design of I-beam section, using the inverted yoke type of steering knuckle, and it is interesting to note that the weight of the 3-ton axle, of I beam design, proved to be 17 per cent less than that of the rectangular Elliot knuckle type of axle formerly used, the stresses remaining the same in both cases. This decrease in weight was obtained in spite of the fact that the I-beam axle was of wider gage than the old type.

In the earlier experimental front axles the steering-knuckle pins were straight where they passed through the axle stock proper, and we experienced considerable trouble due to the straight pins "belling" out the ends of the hole in the axle. This wear was, of course, in the

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more expensive piece of the two, and could not be corrected after it took place without re-machining the axle and putting in bushings, in which in turn the same trouble would occur.

This difficulty was entirely overcome by fitting the steering-knuckle pin into the axle end on a long taper, holding it tightly in place with a nut and sleeve at the lower end, as shown in Fig. 1. This practice, in service, proved so satisfactory that it was used for the torque-arm rear end pin, where it passes through a heavy lug

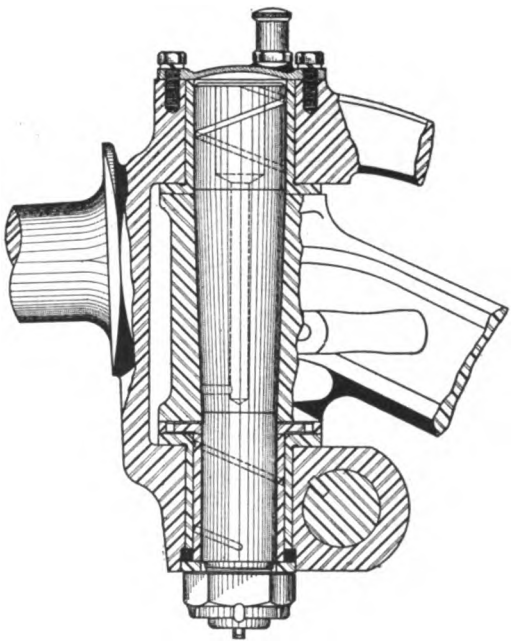

FIG. 1—TAPERED STEERING-KNUCKLE PIN

on the rear axle cast-steel center housing, a point where wear, due to the "belling" out of the hole, would eventually involve an expensive replacement charge.

On a heavy truck it is difficult to make a substantial connection between the steering-knuckle and the steering-arm, for the steering-gear connecting-rod, this being the arm which carries the entire steering effort. We tried pressed hexagon fits, brazed hexagon fits, and pressed cylindrical fits, with four-key dowels, but all these constructions gave more or less trouble, and we finally forged this one lever integral with the steering-knuckle, making the lever a heavy oval section.

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Prior to our experimental work on worm-driven trucks, we had experienced some trouble with our steering connection joints, which were of the clevis type. As originally designed, the pins were prevented from turning in the clevises by the engagement of a flatted side of the pin head with a machined lug on the steering-rod clevis, and the lever eyes were bushed with bronze. The bronze bushings pounded out; so hardened steel bushings were substituted. Then we found the pins pounding out and over-sizing their holes in the clevises. The cross tube and connecting-rod pounded and wore the

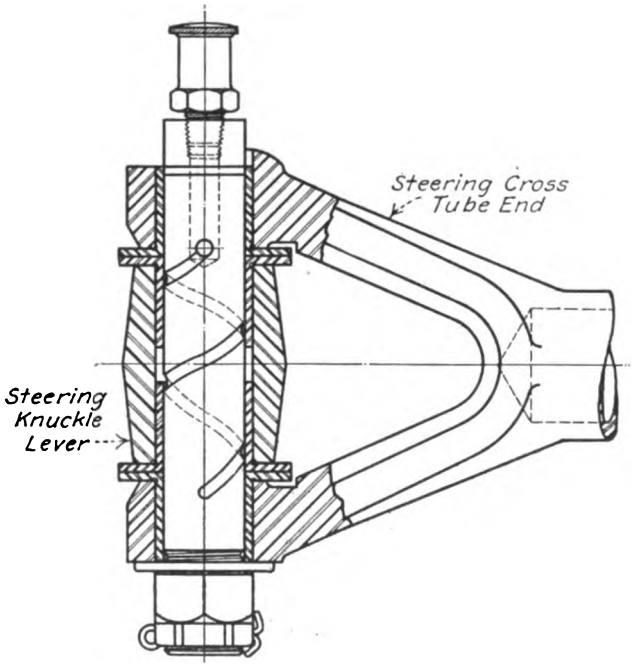

FIG. 2—STEERING-CONNECTION JOINT CONSTRUCTION

faces of the eyes and clevises. We then commenced using hardened-steel collar or "plug hat" bushings in the steering-lever eyes, and in both sides of each clevis. These bushings were so arranged that the weight was carried on large diameter hardened surfaces, as shown in Fig. 2. This construction was only fairly satisfactory, as the continual pounding from solid tires, coupled with inability to retain lubricant, was the cause of too rapid wear.

We felt that we had exhausted all the possibilities of the pin and clevis type of joint, and turned to the ball-and-socket joint construction, using hardened and buffed balls and sockets, no springs being employed behind any of the sockets. This construction was an improvement over the final bushed eye and clevis design, as shown

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in Fig. 2, but still wore and rattled. The wear was obviously due to the slight clearance between the bearing surfaces, through which the vibration resolved itself in a succession of light hammer-blows, the intensity of which increased with wear.

We next tried placing a spring behind one of the ball sockets in each joint, in order always to retain all three parts in close contact. Some experimenting was necessary to determine on a spring pressure heavy enough to allow only a slight deflection under steering effort and road shocks, in conjunction with balls of a size such that the unit pressures could be kept low enough not to cause hard steering. This construction is shown in Fig. 3.

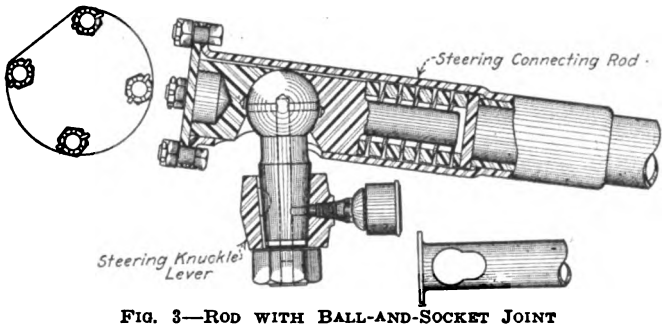

FIG. 3—ROD WITH BALL-AND-SOCKET JOINT

Excellent results were finally obtained on a 3-ton truck with ball of 1⅝ in. diameter, in conjunction with a spring compressed to 300 lb. The balls and sockets are made of steel and hardened. The author has measured balls after 30,000 miles of service that showed less than 0.003 in. wear on the diameter, clearly indicating how slowly the construction wears.

REAR AXLES

In 1911, when we started experimental work on worm-driven rear axles, very little information on worm gearing for truck rear axle work was available in this country, and our decision to develop an axle of this type was based largely on foreign practice. We investigated carefully the construction and service results of internal gear and worm-driven axles in Europe, and were deeply impressed by the wonderful results obtained with worm gearing on the chassis of the Dennis and the London General Omnibus companies.

At that time the Hotchkiss drive looked promising, and that construction was embodied in our first worm-driven vehicle. On a heavy vehicle we could not make the springs as durable with the Hotchkiss-drive as with the torque-arm and distance-rod construction, and after exhaustively testing a model with distance-rods and no torque-arm, which resulted in an objectionable chatter when using the

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transmission brake, we finally decided upon the conventional type of construction, using both distance-rods and torque-arm.

Right at the start we began to have trouble with the axle drive-shafts twisting off at the outer ends. European practice is to use a rigid connection at this point, between the drive-shaft and the wheel hub, chiefly in connection with a floating bronze bushing for the rear wheel bearing. With the taper roller bearings, which are largely used in wheel hubs in this country, a slight amount of play is necessary in the bearings, which permits the wheel to rock slightly. This slight rocking of the wheel, in conjunction with a rigid connection between the driving shaft and the wheel hub, subjects the shaft to bending, and we found that these bending loads, in our case, were the cause of shaft failure.

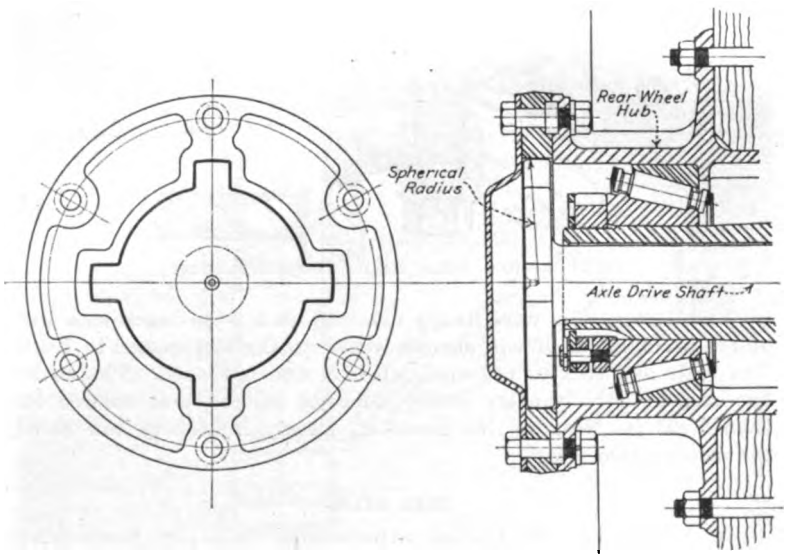

FIG. 4—CONNECTION BETWEEN DRIVING SHAFT AND WHEEL HUB

For the rigid connection between the shaft and the wheel hub, we substituted a four-lug semi-universal construction, which permits a slight rocking of the rear wheel without placing any bending loads on the shaft, as shown in Fig. 4. The driving plate, which attaches to the hub, is hardened, and the heat-treated drive-shaft is also hard, so that the driving loads are taken through hardened surfaces. The pressures on these driving surfaces are as follows:

On low gear.....4230 lb. per sq. in.
On high gear.....1290 lb. per sq. in.

The wear in normal service should not exceed 0.004 in. in 30,000 miles. Since using this construction we have, so far as the author is

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aware, never had a failure at the outer end of the shaft, either experimentally or on any of the thousands of trucks in service.

We also had trouble at the inner end of the axle shafts, where they engage the differential-gear hubs, these failures occurring on trucks in extremely heavy service, which were fitted with high reduction worm gears. The shafts twisted off where the six splines on the inner end run out into the round of the shaft, and practically at the end of the differential gear hub. These axle shafts were made of a nickel-chromium steel, heat treated to the following physical properties:

Tensile strength	155,000 lb. per sq. in.
Elastic limit	120,000 lb. per sq. in.
Elongation	16 per cent.
Reduction of area	48 per cent.
Hardness (Scleroscope)	50 per cent.

On the 3-ton truck, with high reduction worm gears, these shafts were stressed as follows:

On low gear	26,400 lb. per sq. in.
Due to foot brakes*	63,400 lb. per sq. in.

And it is interesting to note that in every case of failure the shafts were twisted off from the low gear loads, and not from the transmission brake loads.

In order to overcome this twisting off of shafts, we made slight changes in the chemical composition of the steel, and changed the heat treatment to obtain the following physical properties:

Tensile strength	185,000 lb. per sq. in.
Elastic limit	175,000 lb. per sq. in.
Elongation	11 per cent.
Reduction of area	50 per cent.
Hardness (Scleroscope)	64 per cent.

To obtain these properties two heat treatments are required. The only way high stresses in the shafts can be avoided is to increase the shaft diameter materially, and such a proceeding is not desirable, as the weight of the entire axle structure builds up around the drive-shaft diameter.

On our experimental worm-driven axles we experienced some trouble with overheating and subsequent failure of the double ball-thrust bearing behind the worm. By building up a dummy housing with cutout sections on top, and driving the worm by belt, we found that practically all the surplus oil carried up to the worm by the wheel

*Stresses for foot brake determined by using coefficient of friction of 0.6 between tires and road.

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impinged against the front annular ball bearing on the worm shaft and that very little of it was forced into the thrust bearing and the annular bearing at the rear.

A cast oil-trough inside the housing was provided to catch the oil thrown off the worm, and from the trough the oil was led through a cored passage into the thrust-bearing housing and allowed to find its way back into the axle case through the worm-gear annular bearing. This practice provides a gravity circulatory oil feed for both rear bearings, and has been found, in service, to insure adequate lubrication.

EFFICIENCY OF FINAL DRIVES

There has been some discussion recently concerning the efficiency of worm gearing, as against that of the double side chain or the internal gear final drive. If a side chain or internal gear final drive is new, and properly lubricated, then under conditions of low rotative speed and heavy torque, the worm drive is less efficient. However, neither side chains nor internal gears can be kept lubricated, and are consequently subject to rapid wear and deterioration, while the worm drive, being completely enclosed, and running at all times in a bath of oil, is just as efficient after 30,000 miles or more of service as it is when new.

It has been the writer's experience that the truck owner is very much more interested in obtaining the minimum cost per ton mile, than in any theoretical discussion of efficiency. In considering the various forms of final drive many factors are of greater importance than the efficiency. Minimum maintenance cost, durability, silence, freedom from the necessity of frequent attention, are all of more importance than efficiency, when the efficiency differences are so slight that they cannot be measured in gasoline or oil consumption.

While the manufacturing cost of a worm drive is greater than that of side chain or internal gear drives, the fundamental reason for its wide-spread use and success lies in the fact that over a given period of time and mileage, its maintenance cost is materially less than that of either of the other two forms.

SPRING-SHACKLES AND PINS

These parts have been the source of a great deal of trouble in the past, because of improper lubrication and wear. The writer has never seen any grease-cup having sufficient power to force grease into the loaded side of the bearing, even with no load on the truck. Grease can be forced in by using a special gun, but such a procedure, while it might be satisfactory for the owner of a fleet of trucks, is hardly feasible for the owner of one or two vehicles. When grease-cups are used in conjunction with grease grooves in the pin, these grooves can be filled up with grease, but further screwing in of the grease-cup only squeezes the grease out on the non-loaded side of the

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bearing, or by the grease-cup plunger or cap. It has been our experience that grease in such grooves is never properly distributed over the loaded surfaces, as the angular motion between surfaces at this point is too slight.

This lubrication difficulty can be overcome by using oil instead of grease, feeding it into a groove along the top side of the shackle pin, whether it is the loaded side or not. The groove should be shorter than the spring bushing, and the oil-cup so arranged as to give a slight head above the groove, as shown in Fig. 5.

When the oil-cup is filled, the truck is stationary and the bushing acts as a seal over the oil groove. In motion, road vibration lifts the bushing slightly away from the pin and permits the oil to flow over both sides of the groove and around the entire pin to the bottom, or in the case of a shackle pin, where the load is normally on the bottom, the oil runs over the sides of the groove and flows down around the entire pin. If desirable, felt can be used in the communicating

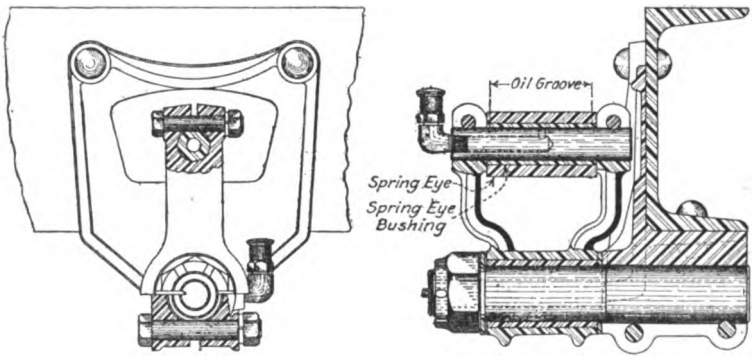

FIG. 5—SHOWING METHOD OF LUBRICATING SHACKLE PIN

hole between the oil-cup and the oil groove, in order to cause gradual flow of oil. This system has proved very satisfactory in service, one filling of the oil-cups in the morning providing sufficient lubrication for a day's run of the truck.

The author's experience has been that even if the shackle pin is a good fit in the shackle, and is held either by a key or a flatted lug against any rotative motion in the shackle, it cannot be prevented from wearing the holes in the shackles out of round. In time this condition also partly cuts the pin in two, and after much of this kind of wear has taken place, replacement of both the shackle and the pin is necessary; and removal of the pin for replacement is very difficult. Working on the assumption that the wear is caused by countless little hammer-blows, owing to inability to make the pin initially a tight enough drive-fit in the shackle, the construction was

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changed to clamp each end of the pin in the shackle, the clamp bolt being notched into the pin and so arranged as to hold the two surfaces tightly in contact. This construction, shown in Fig. 4, has been the means of obviating entirely any trouble due to wear of the shackle or the shackle pin where they come in contact.

RADIATOR AND HOOD MOUNTING

The author has experimented with several different radiator mountings—rubber cushions, coil springs, and C-shaped flat springs. The results of these experiments showed a rigid mounting to be impractical, and that flexibility in mounting is more essential than extreme cushioning. Truck radiators are more frequently injured by the strains put upon them from chassis frame distortion, than from vibration. The most desirable mounting is one that permits frame distortion without straining the radiator assembly and that, while giving a certain amount of cushion effect, is still rigid enough to prevent excessive vibration of the entire radiator on its mounting.

It has been the writer's experience that it is impracticable to carry the front end of the hood on a ledge which is integral with the radiator casing. If the radiator is carried on a semi-flexible mounting, it has a slight vibratory motion, relative to the chassis, which results in destroying the hood hinges and tearing out the rivets at the front end.

We are now carrying the front end of the hood on an angle-iron frame which is bolted to the chassis frame, just back of the radiator. This frame is entirely independent of the radiator and carries a water drip molding to prevent the entrance of water between it and the radiator. The construction is simple and cheap, and has quite overcome the trouble we have experienced in the past, due to the hood rubbing and chafing its supporting ledge entirely off the radiator casing, and destroying itself and its hinges in the process.

The subject of radiators naturally brings up the fan-belt question. A fan-belt is one of the most troublesome little things with which a truck designer has to contend. Of course, in truck practice, the conditions under which the fan-belt must operate are unusually severe, as the fan speed is fairly high and the pulley center distance is short. We run our fan at approximately 1400 to 1000 r.p.m. of the engine, and have never yet found a belt which, running on 14-in. centers, has what we consider sufficient life. We have tested many different kinds of belts, both flat and V-section, but have found nothing better than the riveted segmental V-belt, and that is not entirely satisfactory. It is to be hoped that some of the belt manufacturers will develop in the near future a fan-belt which will give at least 100 per cent more life than the types at present available.

GOVERNORS ARE ADVISABLE

All our experimental and regular service data point to the advisability of using a governor to limit the maximum engine and truck

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speed. It is the author's opinion that chassis depreciation increases almost directly as the square of the road speed, but it is manifestly illogical to try to govern all trucks of the same capacity, to run at the same maximum speed, irrespective of operating conditions. For instance, a truck which is governed to the best speed for proper life and maintenance costs for operation over the rough streets of Pittsburgh ought to be capable of running faster with no greater depreciation over level asphalt streets in Detroit.

As truck engine and chassis design improves, higher engine and road speeds are going to be permissible, and the increase will probably be obtained by speeding up the engine and possibly making it smaller, using somewhat higher gear ratios in order to obtain the present activity factors on direct drive. Betterment of city road conditions will also help to make higher road speeds allowable. The

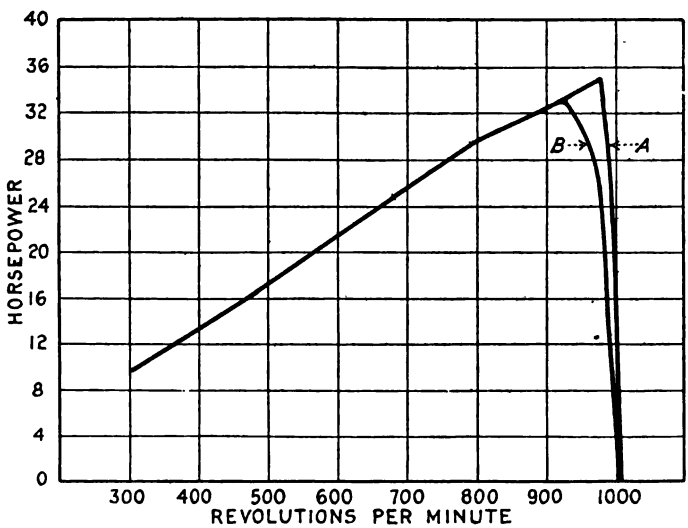

FIG. 6—EFFECT OF GOVERNOR OPENING AND CLOSING
Curve *A*, Action Going Up. Curve *B*, Action Going Down

writer has carried on a great deal of experimental work on centrifugal governors in order to obtain the minimum speed range for governor action without "surging" or "hunting." Assuming maximum engine governed speed is 1000 r.p.m., the governor should not cut into the horsepower curve any earlier than necessary. Our earlier governors used to start acting at 925 r.p.m., closing the throttle tight at 1000 r.p.m. After a great deal of experimental work, we have developed a governor which starts to close the throttle at 950 to 975 r.p.m., and completely closes it at 1010 r.p.m., giving a range of 6.4 to 3.4 per cent. With the engine running at 1000 r.p.m. under light loads the governor should open the throttle as quickly as possible when the engine speed decreases from an increase in the load;

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but owing to the friction to be overcome in starting governor action the governor will not open within as close a range as it closes the throttle. This difference in opening and closing action is shown on the curve in Fig. 6. The curve was made from a 4½ by 5½-in. engine, the loads being checked in speed increments of 25 r.p.m. at all speeds above 900 r.p.m. The governor, with the characteristics shown, does not hunt or surge on the road, and has a "snappy" action, opening or closing quickly, according to variations in the road resistance.

UNSPRUNG WEIGHT

There has been considerable discussion recently concerning sprung versus unsprung weights on trucks, with particular reference, of course, to the rear axle construction. While the author has as yet no conclusive data on the subject, it is his opinion that within reasonable limits, unsprung weight on a truck is not a disadvantage. Unsprung weight on solid tires, at comparatively slow speeds, is not analogous to unsprung weight on pneumatic tires at high speeds. We once made a visual check on contact between the tires and the road, by running a worm-driven and a chain-driven truck at the same speed over a stretch of rough pavement, making the test with the trucks both loaded and unloaded. When loaded, both trucks carried the same weight on the rear tires, but when unloaded, the worm-driven truck was slightly heavier in the rear. So far as could be judged by careful observation, the tires of the worm-driven truck left the road fewer times than those of the chain-driven truck, either when carrying a load or running light. This condition would indicate greater tire mileage for the worm-driven vehicle, as it is obvious that when the engine is pulling, tire wear must be increased when a tire leaves the road and accelerates while in the air owing to the differential, and is suddenly brought back to its original speed by slippage against the road surface upon its return to contact.

The rear axle unsprung weights of the two vehicles tested were 2162 lb. for the worm-driven, and 1612 lb. for the chain-driven, both trucks being of 3-ton capacity. The side-chain-driven truck admittedly results in the lightest unsprung weight construction. The increase in unsprung weight of the worm-driven axle, over the chain-driven axle, is only 550 lb., or 275 lb. per wheel.

DISCUSSION

JOHN YOUNGER:—It is nearly six years since worm-driven trucks were introduced into this country by the Pierce-Arrow Motor Car Co. against really terrific opposition. As was natural, at the beginning we had our troubles. Axle-shaft breakage occurred occasionally in our early trucks, and the means taken to cure it have been followed out to the letter by Mr. Church. This is fully described in a paper I presented to the American Society of Mechanical Engineers in June, 1915.

It is gratifying to find that our early work on I-beam drop-forged front-axles for heavy trucks has met with approval and that the

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tapered steering-knuckle pin has been found good practice. Mr. Church's improvement on the pivot arm, by forging the steering lever integrally, seems promising and may be another solution of the steering-lever connection problem. I think, however, that this arm must be in the way of the machines in finishing the yoke end of the pivot axle.

The lubrication of shackle-pins and other slightly rotative parts is always a problem. Grease does not always go where it ought to, and oil-cups have a nasty habit of collecting grit, which is carried into the lubricant by the oil-can spout and thence into the bearings. Besides, drivers neglect to grease or oil their bearings. I look forward some day to a modified Fergus-car construction being applied.

Mr. Church states that at low rotative speed and heavy torque, the worm drive is less efficient than side chains or internal gears. Particulars should be forthcoming of the actual loads, speeds and all factors entering into the test from which such conclusions were drawn. I cannot quote comparative tests, but will give an example showing the efficiencies of worm gearing at low speeds and high torques.

Near Buffalo is a hill 1413 ft. long, 8 per cent grade throughout, and with a road surface of vitrified brick laid last Fall. Our standard 5-ton truck climbs this on second gear (reduction 16.53) with a small margin of power to spare.

The engine of this truck on the bench gives 45 hp. at 1000 r.p.m. The rear-wheel driving diameter on new tires is 41 in. The weight of the complete truck, with its test load, is 20,300 lb. The resistance of the road-bed is low, only about 10 lb. per ton. (According to published figures this is often 15 lb. per ton, but I believe 10 lb., or even slightly less, is nearer the mark.) Taking the formula

$$e = \frac{n \times \frac{D}{2} \times W\left(\frac{20g + R}{2000}\right)}{63{,}000\ Pr}$$

$$e = \frac{1000 \times 20.5 \times 20{,}300 \left(\frac{160 + 10}{2000}\right)}{63{,}000 \times 45 \times 16.53}$$

whence e, the efficiency of transmission, is 75.6 per cent. That is, the total loss in efficiency due to transmission through two pairs of universal joints, two pairs of gears in indirect drive, worm gear, differential bearings, road-wheel bearings and to deformation of rubber is less than 25 per cent. When we consider that the engine was fitted with a governor limiting the speed to 950 r.p.m. under a full forty-odd horsepower, the high efficiency of the worm gear under low rotative speeds and heavy torque becomes apparent.

The hill used is a permanent institution; its road surface is unimpaired by weather conditions, and the test can be duplicated at any time. Incidentally, the test also proves a governor is not detrimental in hill-climbing.

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AUTOMOBILE EXPERIENCES IN THE GREAT WAR

By W. F. Bradley

(Non-Member)

Abstract

The author outlines the constructions that have performed cially that four-cylinder engines carried under a hood are the most satisfactory. The defects revealed by war service are given in considerable detail, the author finding that all of the trucks used had developed some weak point. Radiators and springs are specified as a general source of trouble. The author outlines a number of operating troubles developed under the existing conditions of operation and gives examples of the way these have been remedied.

Considerable attention is paid to the methods of operating trucks away from made roads. The methods of fitting chains to the wheels, and the use of caterpillar attachments are described. Dimensions are given for bodies and a number of suggestions made as to their proper construction.

Although practically all the general transportation is done by rear-driven trucks the four-wheel-driven vehicle is used to a limited extent, mainly for operation off the main roads or on no roads at all. A description is given of tractors developed for this service and of the trailers now being used behind all kinds of automobiles, both for the transportation of men working in the rear of the lines and for general haulage work around the depots.

In conclusion the author considers the arguments heard in the war zone in favor of standardization and gives in detail a number of fit subjects for standardization. He also mentions the necessity for uniform nomenclature inasmuch as a great deal of confusion has been created by the difference in the names of American truck parts, not to mention the trouble caused by the variation in English and French terms.

BEFORE THE WAR

Before August, 1914, it was realized that automobiles would play a prominent role if the nations of Europe went to war. In the annual maneuvers some nations had gone so far as to abolish the use of horses in the supplying of food and ammunition to an entire army corps, and had found the experiment to be successful. But no single expert, either military or automobile, appears to have foreseen the extensive use of automobiles to be developed within a month after the outbreak of war. In all their preparations, it is certain that the war authorities had never contemplated having to go outside Europe for their supply of automobiles. This contingency was so little foreseen that it is doubtful if the military automobile authorities of France and England could have named the six leading auto-

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mobile truck makers in the United States, or if any of them could have given, from memory, the characteristics of a leading American truck.

Yet the war had not been in progress more than a month before France, the nation that had most carefully studied the use of automobiles in war, had sent an official to the United States with instructions to buy several thousand trucks; England with a greater number of commercial vehicles than any other nation, was also anxious to supplement her military fleet with trucks from across the Atlantic. The trucks available were ordinary commercial vehicles, built for commercial conditions in the United States, without their designers or manufacturers ever having conceived that they might some day be employed in war service. These purely commercial trucks were put to work side by side with the more specialized European machines, and although they were open to improvement in some or many respects, they gave—and are still giving—good service in the war zone.

PART PLAYED BY SUBSIDY VEHICLES

Before the war the British authorities had designed a subsidy type of truck. Admittedly a good vehicle for military purposes, it had been designed with such disdain of commercial service that few business houses would purchase it. When England went to war less than fifty of these army subsidy trucks were to be found in the whole of Great Britain. Then the requisition officers gathered in practically everything on wheels. It was a jump from one extreme to the other, for many of these hastily requisitioned trucks never got within sound of the guns.

The French system was much less drastic. The military authorities, instead of saying dogmatically what were and what were not good features of army truck design, left these matters largely to the engineer and were content principally with imposing working conditions. If a truck could fulfill those conditions, it would be accepted as an army subsidy vehicle, no matter whether it had worm or chain drive, whether its motive power was obtained from steam, gasoline, electricity, or from a combination of these. In the end this tended towards uniformity, for it automatically eliminated unsuitable types. But it did not bar a manufacturer who failed to see eye to eye with the military expert on the question of final drive, or who refused to give up his preference for a particular type of wheel bearing or gearbox.

The problem was, and ever will be, one of directing design into such channels that the whole of the nation's commercial trucks could be applied to purposes of war. This application must be possible however without interfering with the primary usefulness of the trucks as commercial vehicles. It is a matter of compromise. If the truck is highly specialized for some class of commercial work, it will probably not be of great value on war duty; if it is designed only to meet mili-

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tary requirements, the ordinary customer will refuse to buy it, either with or without the offer of a subsidy.

PROPER CAPACITIES

For instance, in France it was considered that the most suitable military type was a 2-ton truck. But the commercial user was not satisfied with a 2-tonner and would have preferred a vehicle with a 5-ton load capacity. Thus a compromise was struck with 3½ tons, and it is this capacity of truck that has proved most suitable under actual war conditions. It is not so light as to be constantly threatened with overloading; it is capable of hauling one or more trailers; and its dead weight is sufficiently low to enable it to operate over poor road surfaces. For the general conditions of war service the 5-ton truck has too great a dead weight.

For certain special classes of work it has been found that a 4-ton truck gives the most efficient service. This model has not been produced in great quantities in Europe, and, curiously, is not found in America. The demand for a 4-tonner is comparatively recent; it does not alter the general statement that the bulk of the work is best done by the 3½-ton vehicles.

Light weight, consistent with strength, has been shown to be essential. The French truck, as it had been developed up to the eve of war, had to weigh 3½ tons (7716 lb.) or less, this including the entire truck with its body, water, gasoline, oil and hoops capable of receiving a suspended load of 2645 lb. In reality the weight was nearly always below the maximum, some of the trucks weighing as low as 5730 lb. and the average being just under 6200 lb. The tendency since the war has been toward a further reduction in weight.

FOUR-CYLINDER ENGINES SATISFACTORY

Four-cylinder engines are insisted on, and these have been found quite sufficient for all ordinary purposes. Some of the heavy tractor engines have six cylinders, but their number is small. The French military authorities have never made any attempt to impose cylinder dimensions. It was in the interests of the manufacturers to get the smallest engines capable of doing the work, for fuel consumption was taken into consideration. The small engines of two or three years ago have however been found unsatisfactory. Moreover the requirement that each truck should be capable of hauling a fully loaded companion up a macadamized 8-per-cent gradient, has tended toward a slight increase in piston displacements, and also toward certain modifications in gear ratios. Statistics of 18 different makes of French army type trucks that went into service August, 1914, show that the average bore was 3.9 in. and the average stroke 5.7 in. The biggest engine measured 4.9 by 6.69 in. and the smallest 3.5 by 5.5 in. bore and stroke, respectively.

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All engines are carried under a hood. There has been a tendency toward this for several years; but the war has shown its advantage over the under-the-seat position so clearly that the latter will no longer be accepted. There are a few exceptions, which only tend to prove the rule, notable among them being the Paris omnibus type of chassis. These vehicles however are used exclusively for carrying troops. On such work the greatest possible body area is required.

The internal-combustion engine has proved itself without a rival for military transportation. No gasoline-electric systems have proved satisfactory. The tendency is toward a reduction in the number of steamers. Even the British, who have always shown a partiality toward steam vehicles, are making comparatively little use of them in France, although steam tractors are much in evidence around the camps in England.

MANY TYPES OF FINAL DRIVE

Under the French military regulations every type of final drive was admitted; the English military subsidy specification called for worm drive. At present automobile trucks are in service on the front with straight bevel drive, double reduction bevel, worm, internal gears, and side chains with and without housings. No one type has shown itself incontestably superior to all the others, and so far as it is possible to estimate their opinion, the higher authorities have no intention of imposing the adoption of any particular type of final drive. The automobile department of the French army has carefully compiled statistics dealing with parts supplied to trucks in service, but it is likely they will be used only to determine the makes that have been most economical and satisfactory, and not which type of design is superior.

While the question of final drive will remain unsolved by the war, it being left to commercial users to work out this problem, the author is of the opinion that chain drive will gradually disappear before worm, internal gear, and double reduction bevel axles. In the French army the White, Packard, Berliet, Clement-Bayard, Rochet-Schneider, Saurer, Fiat and Delahaye trucks are chain-driven. These have given as good service as any others, but the impression remains that the chain is less efficient mechanically than some other types of drive; its efficiency is certainly less under some conditions of operation. The ease with which the chain can be repaired or replaced is an asset in its favor; but the claim of high clearance is open to question. Under certain conditions of operation trucks with unprotected side-chains constantly plow through a bed of mud, and although they can get through it is at the expense of their chains.

DEFECTS REVEALED BY WAR SERVICE

It is not an easy matter to generalize on defects revealed by war service. Not a make of truck but has not developed some weak point

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of minor or major importance, although few of the defects are applicable to more than one or two makes. For instance, one American worm-driven truck cracked its axle housings; this was a case of faulty design, and was a particular and not a general defect. A light American truck literally ate up differential drive-shafts; this was traced to poor material. A French chain-driven truck warped its differential casing under heavy braking stresses. Another had cast timing-gears mounted on bearings of insufficient size. Vibration ensued and the gears were reduced to what the mechanics termed "marmalade." A more general complaint was the breakage of canvas and leather universal joints, it being not uncommon to see drive-shafts flung across the road during heavy pulling in sand and mud.

Spring troubles were fairly general, but in only a few makes did the breakage of a front spring entail the smashing of the crank chamber. In these makes when the frame dropped the basechamber came in contact with the axle. Also, spring failures were almost invariably confined to the front, it being rare for a rear spring to break, even on trucks and cars known to be of poor quality. On trucks this can be attributed to the narrowness of the front wheels; these would drop into small holes over which the rear tires would ride safely, thus putting additional stresses on the front. Few trucks in war service had sufficiently heavy front springs. The generally adopted method of attaching the springs to the axle cannot be considered safe. Not one driver in a hundred thinks it necessary to give any attention to his spring clips, yet these have a habit of working loose, and in many cases when an axle has shifted it is practically impossible to steer the vehicle. Frequently the complete breakage of a front spring is to be preferred to the loss of nuts or the breakage of clips.

Radiators were a general source of trouble, many a good engine being burned out by reason of small diameter tubes or by brackish water. The honeycomb radiator is not suitable for army truck service. The large diameter vertical-tube type is most successful, and is adopted whenever replacements become necessary. It has been found to give good service even when used on trucks originally equipped with the honeycomb radiator. Another good type is the circular tube radiator with a centrifugal fan, as used on the Paris omnibus type of chassis.

All army trucks should have radiator guards attached, not to the radiator itself, but to the frame members. The design should be such as not to interfere with free operation of the starting-crank. The guard should be so designed too that it will not allow the overhanging rear of a truck body to pass above it. In some cases it has been found advisable to fit a transverse bumper with coil springs at the rear of the trucks, this bumper coming in contact with the center of the radiator guard in case of a collision.

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REQUIREMENTS IN ENGINE DESIGN

Considering a 3½-ton truck as the most suitable type (and of this all who have had experience in the war zone are in agreement) the engine should be a four-cylinder L-head of about 450 cu. in. piston displacement. It has already been stated that the under-the-seat position is not suitable except for special service where a maximum body space is required. The underpan is an abomination on an army truck. The basechamber should be easily removable, although unfortunately this does not necessarily follow when there is no mudpan. The connecting-rod bearings should be detachable, so that it is possible to change them, in case of necessity, without taking the engine out of the frame. A governor is a desirable equipment.

War service has tended to prove the superiority of pump circulation over thermosyphon. As a general rule trucks tend to overheat when operating in convoy formation on crowded roads. This condition, only occasionally met with in civilian service, is common on certain portions of the front, and should be provided for. There are few trucks on which sufficient attention is paid to draining off all the water from radiator, pump and jackets. This is a detail when trucks are housed in a heated garage; it is an important feature when automobiles remain constantly in the open air, with the possibility that the engines may have to be cranked, to meet an urgent call, at 2 or 3 o'clock in the morning. No man who has had to start an engine at daybreak on a frosty morning, with the knowledge that delay meant disaster, is inclined to treat this matter lightly. It is common practice for drivers to make straw mats to fit the hood and the radiator, and to use these as heat retainers whenever the truck has to stand idle for a few hours. An additional precaution is to disconnect the gasoline line, empty the float chamber of the carbureter, and leave one of the kerosene side lamps burning under the hood. The manufacturer might help in this direction by making a tightly fitting hood with louvers that could be opened or closed at will, by supplying a heat retaining radiator cover and also providing a safety lamp capable of maintaining a certain amount of heat under the hood when the truck is left standing in the open all night. An adjustable air-shutter on the carbureter is a valuable fitment. It is important too that the engine should develop something like its full power within a few minutes of starting.

Motor starters are not necessary on army trucks. In the few cases where these have been fitted the batteries have generally been requisitioned to light some dug-out or office. As there are always two drivers on army trucks, it is hardly necessary to relieve the spare man of the only real work he has to perform.

Ignition by high-tension magneto has proved all that could be desired for war service, no need having been shown for a second system, either as a reserve or for ease in starting. The only general trouble has been the swelling of the fiber and condensation on the

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spark-plug points, this being due to the excess of moisture in the atmosphere of Northern France. The French military authorities have insisted on the standardization of magneto bases and couplings, and also on all magnetos turning right-handed. The intention, of course, was to make all magnetos interchangeable throughout the army, and although this result has not been obtained, owing to the influx of foreign machines, the system has been good so far as it has been possible to apply it.

LUBRICATION AND FUEL

It has been noticed by officers in charge of army repair depots that engines with the old-fashioned splash system of lubrication are generally better preserved than those with pressure feed of oil to all bearings. On war service the lubricating oil is apt to remain in service longer than is desirable, and the impurities deposited in the oil are picked up by the pump and forced into the bearings. The remedy seems to be, not a return to splash, but a pressure or circulating system so designed that the dirt will deposit away from the pump. It is really not necessary that the pump should be in the lowest portion of the basechamber. The basechamber can be designed so that the lowest portion is merely a trap for impurities, and this can be done without any danger of the pump being accidentally deprived of oil. Whatever the system it ought to be an easy matter to dismount the pump and to drain the basechamber of every drop of lubricant. This presupposes the absence of a mudpan. Several American trucks have suffered materially and in reputation by the absence of a suitable oil. Pressure feed being common in Europe, a heavy oil is generally employed, and this has been used with disastrous results in engines requiring a light bodied oil. The only remedy appears to be for manufacturers to clearly indicate the type of oil to be used on their engines.

So far as the Allied armies are concerned, there has been no necessity to use alternative fuels such as benzol, alcohol and kerosene, although it was a pre-war specification in France that all automobile trucks should operate on benzol and on a 50-per-cent mixture of benzol and alcohol. Benzol and alcohol have been monopolized for making explosives, and gasoline has been the only army fuel in use. It is certain, however, that the benzol requirement will be insisted on after the war, and in all probability alcohol will be a widely used fuel.

The best place for the fuel tank is on the dash. Pressure tanks at the rear are inconvenient and dangerous. The filler should be of sufficient size to allow a man to pass his hand inside the tank, and there should be provision for draining the tank easily. Drivers in the war zone invariably carry a reserve supply of gasoline, this being done because of the feeling of security it imparts rather than because of any urgent necessity for its presence. Suitable provision

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has not been made, particularly on French and American trucks, for the carrying of this reserve supply.

STEERING AND BRAKING

There is an amazing lack of uniformity in the steering lock and the turning radius of army trucks. This is most disadvantageous when convoys are composed of different makes of trucks, for while one vehicle can make a turn easily the one following may be unable to do so. The disadvantage is also felt where makes are grouped, for one convoy may negotiate a difficult road with ease while the next will be stopped. In some makes of trucks the short turning radius is only an illusion, for owing to the radiator extension in front of the axle it is only possible to turn within the given radius if the road is entirely free. In other cases the radiator is so high that it is impossible for the driver to see directly ahead and take full advantage of his lock. This matter is important, for when a road is under fire the greater the time spent in making a turn the greater the chance of being hit by a shell.

Only the southeastern end of the present battle line is in a mountainous district. Thus brakes have not been put to a severe test. Generally there is much room for improvement in the matter of brake adjustment. Several cases have been noted of brake-rods so close to the road or to the road wheel that they have been bent by a board lying in the road and thrown up at an acute angle as the wheels passed over it. Brake-rods should be regarded as organs needing protection and placed in such a position that they are not likely to be damaged by obstacles on the road. Not many American trucks are fitted with a sprag, although this is a valuable accessory on military trucks working in convoy formation in hilly country. Picture a closely packed convoy on a greasy hill, trucks without sprags, and a green driver in the center of the column who misses his gears and fails to use his brakes in time.

Four-speed gearboxes are now insisted on by the French military authorities and are found to be indispensable for active service. First speed should be equivalent to 2 m.p.h. with the engine running at its normal speed (1000 to 1200 r.p.m.). One type of gearbox that has proved satisfactory under war service is the Damaizin patent constant mesh gears. This gearset was fitted to the Paris bus chassis and its use has been extended to other makes. It is impossible for a driver to muddle his gears no matter how ignorant he may be of the elementary methods of changing. No experiments have been made with trucks without a differential, but a differential lock, capable of being operated from the driver's seat, if possible, is an absolute necessity under many conditions of active service. The only reproach brought against one of the best American trucks on the French front is the absence of this lock.

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SECURING TRACTION OFF MADE ROADS

The greatest objection brought against the use of the automobile truck is the rapid fall of its efficiency as road surfaces deteriorate. The touring car, the ordinary commercial truck applied to war service and the various types of four-wheel drive and purely military vehicles are all, in varying degrees, open to improvement in this respect. The general rule is that convoys should keep to made roads and not attempt to operate across country. Officers using touring cars are specifically forbidden to order their drivers to go across country. While this general rule is observed, there are frequent circumstances where the made road practically ceases to be a road, or where the only means to safety lies in a dash across country. The final distribution of food and ammunition under really difficult circumstances is still effected by horse teams or by men. A two-wheel cart with a ½-ton load will get over a road where a 3-ton vehicle, whether drawn by horses or propelled by a gasoline engine, would be in difficulties. If the light cart is hit by a shell the wreckage is easily cleared away and the procession continues; but a heavy truck disabled on a narrow dirt road stops all further traffic.

The most common expedient to render a truck fit to travel away from hard road surfaces is to fit chains to the wheels. These, however, have been far from giving general satisfaction. The circumferential chain, lodged in the space between the dual tires has not been a great success, for the available space is too narrow to allow of a heavy chain being fitted, and breakages are frequent. The same objection applies to the type consisting of a circumferential chain placed between the dual bands, with transverse ribs at regular intervals having one surface in contact with the tread of the tire and the opposite face in contact with the road surface. The chain uniting these ribs has always proved too weak.

Short lengths of chain, hooking from the outer to the inner face of the rim, across the tire, fitted among others to the Pierce-Arrow trucks, are fairly satisfactory. The lengths being independent, the breakage of any one does not affect the efficiency of the device. Experiments were made with the same trucks with a caterpillar attachment secured to a flange bolted to the wheel felloe. This device was quite unsatisfactory. After a truck became bogged it was impossible to apply the attachment, and if used over a made road it broke the chains or bent the steel shoes in such a way that they could not be used a second time. In the case of some of these trucks, carrying a heavy load of armor and an anti-aircraft gun, it was found that if the ordinary chains would not take them through, nothing else would. Thus these chains were always kept on when operating away from roads.

It is surprising, however, how soon a set of chains will be destroyed if used over a hard macadam road or granite surface. In a short time a set of chains will be reduced to individual links and

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portions of links under certain road conditions. Thus the necessity arises for a quickly detachable device, so that it can be taken off immediately a hard road surface is encountered. The ordinary chain devices are not applicable to chain-driven trucks, the clearance generally being insufficient between the road wheel and the sprocket to allow a chain to be attached.

A device adopted as the result of practical experience in the field consists of an endless chain passing round the road wheel (fitted with dual tires) and around a loose pulley some distance back of the road wheel. A spring-controlled bell-crank arrangement operates on the pulley and maintains a sufficient tension on the chain. On greasy roads and mud-coated granite surfaces this is a most effective non-skid device and one that gives little trouble through breakage. It was found to be useless, however, on soft earth. Under these conditions the chain refused to revolve with the driving wheel and the whole thing became embedded.

The problem of securing traction has not been confined to rear-wheel drive trucks. Four-wheel drivers have had to face the same difficulties although of course in a different degree. The most difficult feature has been to find a device that would prove satisfactory under all circumstances.

METAL WHEELS BEING ADOPTED

One of the most important developments of the war is the adoption of the all-metal in place of the wood wheel. Although the wood wheel may continue in use for some years on commercial trucks, and possibly will not be abandoned for touring car purposes, it is already doomed for army trucks. The primary objection brought against the wood wheel is that, even if well made of good material in the first instance, it requires attention from time to time to keep it in proper condition. This defect is most apparent when trucks have to remain out of doors day and night, indeed on all occasions except when in the repair shop. Another objection is that if the vehicle catches fire—and the enemy's guns are constantly on the search for convoys working back of the lines—the wheels are liable to be destroyed and it becomes a difficult matter, if not an impossibility, to get the truck away. With metal wheels it is nearly always possible, when the fire has burned itself out, to tow the vehicle home. Apart from the body, it is surprising how little a burned-out truck will suffer in its essential organs.

Practically no army trucks are now being built in Europe with wood wheels. All the Austin and other trucks supplied to the Royal Naval Air Service for use in France in a short time had their wood wheels changed for those of the steel disk type. Generally when replacements become necessary cast-steel or disk wheels are substituted for wood. A small number of American trucks have been changed in this way.

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For trucks of 3½-ton capacity and upward the cast-steel wheel of the spoke variety is extensively employed by French and Italian makers, and also by the British. The latter, however, are also partisans of the disk type. For the lighter units, such as light trucks on twin pneumatic tires and motor ambulances, the steel-disk wheel has made a remarkable jump into favor. These pneumatic-shod wheels can be detached by removing four to six nuts. Incidentally, four studs and nuts, adopted by some makers, are considered an insufficient attachment. Six should be a minimum. The detachable wire-spoke wheel has not been given a more extended application although it has held its own on touring cars used by staff officers.

RUBBER TIRES USED ON ALL WHEELS

Not many years ago the French military authorities were inclined to favor the use of steel tires, particularly on the driving wheels. The results were not satisfactory, the initial economy being more than offset by the greater cost of upkeep and the difficulty of operating the vehicles on greasy granite-paved roads. The war has proved the necessity of using rubber on all trucks and on all wheels. Block tires, for which many claims were made at one time, have not proved satisfactory. The alleged advantage of being able to replace a block if one gave out has not been substantiated. One reason is that the new block always has a greater thickness than the worn unit it has to replace.

All kinds of tire troubles have developed and moreover kinds unknown under peace conditions. Mileage has dropped very low and many trucks in the north of France are not averaging more than 1500 miles on a set of band tires. On some of the heavy gun cars 800 miles is considered a fair average and 1500 miles excellent service. These, however, are special vehicles having a load of 7 tons and more, obliged to operate in a fire-swept zone not negotiated by any vehicles other than the light two-wheel carts of the Army Service Corps. One section of 5-ton trucks doing ordinary transport service in France averaged 5000 miles on a set of tires, but 4000 miles can be considered a fair average.

It has been suggested that the rubber used during the war is not equal in quality to that of peace times. There is nothing substantial in support of this contention, however. It is true that some damage is done by improper press work. The hydraulic presses to be found in all the repair depots are often in the care of men who cannot be considered experts and who shorten the life of a tire in the fitting process. Tires do not perish principally by having their tread worn down. They chip away laterally; they come away from their base, sometimes for short distances, sometimes for their entire circumference; and generally they wear unevenly. On all the cambered roads of France the inner one of dual tires carries the greater portion of the load and of course wears down before the outer one.

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It is worth suggesting that some means of equalizing wear be devised; a detachable and reversible wheel, separate from the bearing housing, would make it possible to get the full life out of a pair of bands, instead of having to scrap both when only one is worn.

An engineer officer connected with one of the convoys, who was troubled with this uneven wear of tires, often followed behind the trucks on a low sidecar. The road consisted of a centrally paved strip, 10 ft. wide, with the outer strips composed of a mixture of mud and more or less broken stone. When passing other vehicles it was necessary to run with the right-hand wheels in the mud, the left-hand side of the body then being a foot higher than the right-hand side. Under such conditions it was possible to see daylight under the outer of the dual tires, showing that the wear was all on the tread of the inner tire. It was also noticed that the standard equipment for front wheel tires was insufficient for the rough roads near the front. When dropping into shell holes and bumping over obstructions it was common to see chunks of rubber torn out of front tires, causing them to be withdrawn from service more frequently than the rear ones.

The experiment was therefore made of fitting wider front tires and narrower rear ones. In the place of the original set of 36 by 5 front and 40 by 6 dual rear, the engineer had fitted 36 by 6 front and 40 by 5 dual rear. This change was satisfactory; the mileage being greater and the cost less. There was less tendency for the front wheels to bog and drivers said that road shocks at the steering wheel were less. The angle of wear for the inner tire was reduced, the outer one taking the load sooner owing to the narrow tread. The tendency to skid was also considerably reduced.

BODY SIZES AND CONSTRUCTION

The body dimensions fixed for French trucks and confirmed by the war are a platform of 138 by 67 in. with removable sides 24 in. high. The driver's seat must be 55 in. wide. The hoops carrying the canvas top have to be of sufficient strength for each of them to carry a load of 660 lb., the central hoop receiving 1300 lb. This allows wounded men on stretchers to be brought to the rear when trucks are returning after delivering supplies. Owing to this specification the hoops are generally of steel and are fitted with rings.

Many of the American trucks supplied to the Allied armies had a prairie-schooner type of body, with a single top covering the load and the driver. This was a mistake as the driver was left without protection when the nature of the load necessitated the removal of the top. Also with no division behind them, the men on the front seat were in the position of a person in the mouth of a drafty alley. It is essential that the two tops be independent, and that there be a vertical division just back of the driver's seat. Drivers need much more protection than is given on the standard

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commercial vehicle. This protection has not been given by American makers. The drivers have therefore provided it themselves, sometimes with peculiar results. The British have a preference for a permanent driver's cab with side doors and canvas extension above the dash, so that in bad weather only the upper portion of the driver's head is exposed. The roof of the cab carries a rifle rack.

It is a primary condition that the rear wheels must not break the uniformity of the body platform. On this account the body is raised a little above the frame members. Advantage should be taken of the lateral overhang of the body to fit stout lockers alongside the frame members. This has been done in many cases to provide storage space for the regulation spares the truck must carry. Few makers appear to have realized, however, that a couple of men having to live and eat aboard the truck also need spares, and that no space is provided for their storage. This could easily be remedied by fixing additional lockers alongside. The construction of these boxes should be strong and door hinges should be at the top—that is the door should open upward.

Although special vehicles are provided for the rapid transportation of troops, all trucks are liable to be called upon for this work on occasions. It is thus desirable that the body should be so designed that it can receive either a couple of longitudinal seats down the center and about 6 in. apart or transverse seats. The accommodation required is of the most primitive nature, and the fittings should not interfere in any way with the normal service of the truck. The brightly tinted yellow or sandstone canvas tops supplied with many American trucks are a danger under European war conditions. The correct tint is gray, or blue-gray, or better still variegated with blue-gray as the dominant note.

USES OF FOUR-WHEEL DRIVEN TRACTORS

Practically all the ordinary transportation of the Allied armies is done by rear-drive trucks. The four-wheel driver is doing special work and does not usually come into competition with the two-wheel driver. In other words the two-wheel driven truck is the most satisfactory and the most economical type for the general transportation of food, ammunition and men. The four-wheel driver first came into use five or six years ago when the French army sought to abolish the use of horses in the haulage of its heavy field pieces. For this work it was impossible to consider the ordinary truck, for the artillery rarely operates along main roads, and in the majority of cases has to work on no roads at all. What was required, therefore, was a tractor capable of hauling the guns to all positions open to horse teams, and of doing the work faster than it could be done by horses. The abolition of horses for haulage of the guns also implied the abolition of horses for the transportation of munitions; thus the tractors had to be capable of hauling trailers loaded with shells to any position that might be taken by the guns.

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The problem has been a difficult one, for it was not by any means sufficient to duplicate the points of application of the power in order to obtain a vehicle that could travel over any kind of country and mount any kind of obstacle. The difficulties met by ordinary trucks when obliged to abandon main roads also had to be faced by the four-wheel driven tractors. The difference is that the four-wheeler's real work begins when the going is such that the ordinary truck would have ceased to be of any use. A large amount of experience has been acquired in the use of chains, paddles and caterpillar bands for the road wheels in order to allow these tractors to operate under all kinds of cross-country conditions. The results, however, have been sufficiently satisfactory to justify the belief that the horse has ceased to be necessary to the heavy artillery batteries.

TRACTORS DEVELOPED BY FRENCH ARMY

The French army has developed two different types of four-wheel driven tractors: a heavy tractor capable of hauling not less than 12 tons, generally on two trailers; and a light tractor hauling a minimum of 8 tons. The unloaded weight of the heavy tractor is limited to 5 1-3 tons, and the light tractor must not weigh more than 3½ tons; these weights comprising the complete vehicle, two drivers, gasoline, oil, tools and spares. Although some six-cylinder engines were used for this work in the early stages, at present four-cylinder engines are employed exclusively. Every tractor is fitted with a power-driven winch or capstan. This is absolutely essential, war experience having shown that even a four-wheel driver cannot work successfully across country without this auxiliary. As a general rule the winch placed horizontally across the front of the frame has been the most successful type. Towing hooks are fitted front and rear, an artillery-type elastic coupling is provided at the rear, a powerful sprag or ratchet is fitted and the tractors have equal size dual tires front and rear. The use of these tractors has been confined largely to the haulage of 120 and 155-mm. field guns, which are heavy artillery that must nevertheless possess a reasonable amount of mobility.

The four-wheel driven tractor has not been used to any great extent in the haulage of the 75-mm. field piece. Nevertheless a certain number of the 75-mm. batteries have been motorized. The weight of this gun, without its carriage, is only 1014 lb. Thus it can easily be carried on a chassis designed for a 3½-ton load. The problem to be solved was a system of quick-acting jacks that would take all the weight off the springs when the gun went into action and give the same rigidity as with the ordinary gun carriage. For these automobile guns the authorities have selected De Dion Bouton and Schneider 3½-ton chassis similar in general design. The engine is a four-cylinder of roughly 4¼ by 6 in. bore and stroke; the gearbox provides four speeds; and final drive is by internal gears. These batteries have been extensively and successfully employed for anti-aircraft work. The ammunition wagons are of the same general design

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as the gun-carrying chassis. Unlike the ordinary trucks the hood is armor plated. The gasoline tank is placed at the rear inside a steel box without a top. This gives it adequate protection against bullets and shell splinters.

HAULAGE OF HEAVY ARTILLERY

An entirely distinct class of work is the use of automobile tractors for the haulage of heavy artillery—14, 15 and 16-in. guns. For this class of slow-speed heavy haulage, steam has been abandoned in favor of the internal-combustion engine. Although varying considerably in details of design, these tractors are all on the same general lines. They embody little of the automobile type chassis. In one particular case a pure traction-engine chassis, made by a steam traction-engine manufacturer, is employed and is fitted with a six-cylinder Knight engine of about 6 by 6½-in. bore and stroke. The engine operates the large diameter metal road wheels through internal gearing and in addition drives a powerful winding drum. The tractors haul loads of 25 to 30 tons distributed over two or three trailers. Their rate of travel is 3 to 4 m.p.h.

For this work the caterpillar tractor is being used to a moderate extent by all the Allied armies. Except for agricultural purposes Europe had paid little attention to the caterpillar tractor before the war. Thus the machines now used by the armies are practically all of American origin. On certain occasions when the enemy's lines have been broken by heavy artillery fire followed by well-ordered infantry attacks, it has been found difficult to make this temporary success a permanent advance by reason of the slow rate at which the supporting artillery has been brought up over the broken country. It is under such conditions that the caterpillar tractor can be used to best advantage, as it can bring up guns more quickly than they can be hauled by horses.

The necessity has been shown for specially trained crews for the handling of four-wheel driven tractors and of heavy artillery tractors and caterpillars. An ordinary automobile driver is not sufficiently experienced. The men, and the officers in command of them, should know immediately what kind of country can be traversed with and without chains and paddles on the wheels; what hills can be climbed with the full number of trailers; where to drop the load; and where and how to put the winch in operation.

The adoption of motor traction for a heavy howitzer battery means that the entire work of the battery is performed by mechanical traction. Thus a battery with forty-five tractors will have attached to it about forty 5-ton trucks and twenty 3-ton trucks for bringing up ammunition and supplies, as well as about fifteen cars for repair service.

WORK OF ARMORED CARS

Practically everything on wheels capable of carrying a machine gun has been fitted up for service at some time or other and termed an armored car. Nearly two years of fighting is a sufficient length

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of time to allow many errors to be corrected and to give plenty of accurate information on what armored cars can and cannot do. Even now, when the unsuitable types have been eliminated there are several varieties of armored cars. At one end of the scale are the motorcycle and sidecar carrying a machine gun, and at the other end the heavy truck, weighing about 12 tons and carrying four machine guns and one field piece.

It should be pointed out, however, that an exaggerated importance has been given to the armored car. In the present stage of war on the Western front, with men in trenches 10 ft. below the ground and in dug-outs 20 ft. below the earth's surface, there is not much opportunity for a scouting automobile to show its merit. But before this underground warfare was adopted armored cars were important, and they will doubtless again be important when open fighting is resumed.

The most suitable type of machine has been found to be a powerful touring-car type of chassis with a four-cylinder engine of approximately 4 by 6 in. bore and stroke, or its equivalent. The single compartment body is built up of steel plates, generally 0.3 in. thick, capable of resisting rifle fire at close range. The plating extends over the engine and the radiator, with louvers to allow a sufficient draft of air for cooling purposes. Steel disk wheels are fitted, with twin pneumatic tires at the rear and fenders designed to deflect bullets without interfering with the accessibility of the wheels. Double steering, although a decided advantage, has not been adopted in the majority of cases. No ordinary chassis is built to steer from both ends, and rather than undertake considerable work the cars have been sent out with ordinary steering. In practice the double steering is never required except at rare intervals and for short periods. These cars usually carry one or two machine guns in a turret, or sometimes a machine gun and a cannon.

TRAILERS BEHIND ALL KINDS OF CARS

The first application of trailers was in the automobile service, in which loads are bulky and light. Their use has been extended in this service and has also been adopted by other branches of the army. Naturally the four-wheel driven tractor has tended toward a considerable increase in the use of trailers, but these are vehicles originally designed for operating with trailers. The trailers are now being attached behind all kinds of automobiles, which, it was originally thought, would never have to receive them. Thus 3½-ton trucks working with bulky loads are being made more efficient by the addition of trailers. A large number of light trucks, of ¾- and 1-ton capacity, mounted on pneumatic tires, are fitted with the special army type elastic coupling at the rear and will take one or two 2-wheel pneumatic-tired trailers. These are extensively used for the transportation of men working in the rear of the lines. For instance, at a big aviation depot employing 500 men, who have to work five miles from their billets, these light tractors and trailers are used to

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take the men backwards and forwards morning, noon and night. The machines are not kept specially for this purpose, but are used on general haulage work around the depot. The trailers are merely two-wheel floats with canvas tops on detachable hoops; the front is closed and entrance is at the rear; there are two longitudinal seats. In a few minutes the trailers can be stripped to mere platform bodies suitable for carrying aeroplane wings or a complete aeroplane. As a speed of 35 m.p.h. can be maintained on good roads, these outfits are valuable, in case of necessity, for the quick transportation of troops to threatened points.

WAR STANDARDIZATION LESSONS

As already pointed out, the attempt to impose on manufacturers a purely military type is doomed to failure. No army can maintain in peace all the trucks it will require for war conditions. Any military type that can be developed is bound to be swamped by the thousands of purely commercial models that will have to be enrolled when the nation goes to war.

Certain features can be insisted on—and indeed are required in Europe at the present time—without handicapping the designer or making it difficult for him to adopt improvements as they are suggested by experience. The features that should be uniform are: size and style of bodies; wheels and tires (in France these have now been reduced to one size); magneto bases and couplings; carbureter flanges; towing hooks; turning radius; clearances; driving chains; threads for all bolts and nuts; and control.

All these features can be adopted, for trucks of a given capacity, without interfering in any way with the scope of the designer and without arresting progress. Although the army type body may not be suitable for all classes of business, it is an easy matter to make the one adopted interchangeable with it. But it is in the repair shops that the real lack of standardization is felt. The repair staffs that have to handle every kind of vehicle from the automobile factories of the whole world, can be excused for believing that not even the fringe of this problem has been touched. Not only does there appear to be an entire lack of uniformity between one firm and another, but even the product from individual factories is not sufficiently standardized. The military authorities have every reason to encourage the movement towards standardization in automobile engineering.

The archaic system of naming automobile parts is another difficulty that ought to be removed. White has a set of names that are not in concord with those of Packard or Peerless; English terms are not the same as those of America; French terms are more uniform than English, but they are open to improvement. The confusion is so great that in the big stores, where French, American, English and Italian spares are given out, it has been necessary to adopt a new international code. It is one of the curiosities of the war that France should be interested as much as America in the standardization of the technical terms of the American automobile industry.

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MECHANICAL TRANSPORT MOBILIZATION

By Arthur J. Slade

(Member of the Society)

Abstract

The paper opens with a number of quotations from publications issued by the Army War College and showing the bearing of motor transport on a proper military policy for the United States. The author then describes two experimental trips recently made by motor-truck owners near New York in an effort to determine proper motor-transport operating conditions. A statistical summary is given for these two experimental trips.

The Army War College has issued in compliance with instructions of the Secretary of War a "Statement of a Proper Military Policy for the United States," supplemented by a number of pamphlets dealing with particular features of this military policy in considerable detail.

In many of these supplementary pamphlets there appears a considerable amount of material bearing upon the subject of motor transport and from them the brief quotations in the following paragraphs are taken.

MOTOR TRANSPORT IN CAMPAIGN

"It (motor transport) has become absolutely essential to the efficient prosecution of a campaign."

"The history of the present war indicates conclusively that all attempts to employ a special type of car or truck for service have ceased. The type of truck ordinarily in use in the particular theater of operations before hostilities offers the most adaptable and suitable transport for war in that theater."

"Motorcycles have generally proved unsatisfactory, and for messenger and orderly service they have been replaced by the light motor car."

"Closely connected with the use of motor transport comes the importance of good roads, and next the question of bridges and a study of the means to be taken to strengthen the highway bridge ordinarily encountered in this country. Heavy ordnance, together with a continual stream of motor transport, will without doubt test the average highway bridge in many probable areas of operations. The development of heavy ordnance has called for the use of the motor in its transportation. There is plenty of information on hand to show that the transport of heavy ordnance, away from the railroad lines, has been accomplished by special motor vehicles."

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"Why should we not accustom not only the troops but a number of officers and men to the use, handling and repair of motor vehicles?"

"The question is at once asked whether we have taken steps to use this transport and to avail ourselves of the large amount of suitable material existing to-day in the United States. The answer is that this has been done as far as existing appropriations will allow. However, most of these vehicles are operating singly or in pairs, and at no one place are there sufficient for one company."

"As yet no attempt has been made to collect the material for automobile parks or for repair shops and these are shown by the experience of all to be badly needed in field operations."

"It is true we can recruit plenty of men from this class (chauffeurs and mechanicians) when war is imminent, but it is one thing to be a chauffeur and another to be a military chauffeur."

PULLING TRUCK FROM DITCH (FIRST TRIP)

"The Quartermaster General's Office has prepared a provisional plan for utilizing motor transport under existing laws and this plan includes a contract system which will take the place of the prizes and subsidies that have been found so efficacious abroad."

"We have conserved the underlying principle for the use of mechanical-driven transport, and this is that it is a transportation unit pure and simple. It picks up a load at one place and discharges this load upon arrival at destination."

"Notwithstanding the fact that it may be some years before we can use motor trucks in our first and second lines of transportation, the fact remains that, in the event of a war, we will have need for this kind of transportation in great quantities behind our second-line transportation."

RESOURCES IN VARIOUS MEANS OF TRANSPORTATION

"There are now about 140,000 motor trucks in the United States. An army of 1,000,000 men might need about 24,000 motor trucks."

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"Only a small proportion of existing motor vehicles are suitable for military service. It is, therefore, not planned to use every vehicle that may be presented, but rather to accept individuals as chauffeurs and mechanicians, and only such vehicles as may be of an approved type and serviceable."

"Volunteer civilians having special training in handling transportation are to be organized and employed in the Quartermaster Corps."

"The War College Division recommends utilizing these great resources of the Nation in men and material in two ways: First, by forming the various units that make up divisions, and lines of communications, etc.; and second, by commissioning individuals whose personal abilities make them desirable for planning in time of peace and for execution in time of war."

MESS EQUIPMENT IN OPERATION (FIRST TRIP)

POSSIBILITIES OF COMMERCIAL INSTALLATIONS

Many of the other War College pamphlets, such as those on "The Development of Large Caliber Mobile Artillery and Machine Guns in the Present European War," "Fortifications," "Mobilization of Industries and Utilization of The Commercial and Industrial Resources of the Country for War Purposes in Emergency," "Organization, Training and Mobilization of a Reserve for the Regular Army," and "The Coordination of the Mobile and Coast Artillery Units of the Army in the National Defense," suggest again and again that existing motor-truck installations employed in commercial-merchandise transport can advantageously be organized both as to material and personnel and become instantly available when necessity arises for the transportation of troops, supplies, artillery, ammunition, material for

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construction of defensive fortifications, material for construction and repair of military roads and bridges and for engineering works.

A number of owners and operators of motor trucks ranging from an installation of one vehicle up to fleets of several hundred machines have interested themselves in the conduct of experiments and test runs with a view to placing at the disposal of the War Department, not only their experience and personal services as motor transportation experts along commercial lines in times of peace, but also such data and information as can be secured by experiments in operating companies of motor trucks in convoy, determining the number of officers, orderlies, drivers and mechanics, the emergency material necessary to be carried for the temporary repair of roads and bridges, the quantity and character of repair parts for the trucks and the commissary and quartermaster's equipment and supplies necessary to make each transport company an entirely self-sustaining unit.

HOMEWARD BOUND ON FIRST TRIP

RESULTS OF EXPERIMENTAL TRIPS

Two experimental trips have been made by two different groups of motor truck owners. These trips have been made with the full approval of the Department of the East and of the National Guard Division of New York.

The first run was planned for a date early in April; the trucks were to start after they had completed their ordinary half day's Saturday work in commercial service, run to the camping point in Rockland County, New York, spend the night and return to the starting point the next day. When the day of the start arrived it was raining and snowing in the city and was snowing steadily in the country.

Before starting a typewritten order of the day, corresponding to orders of the day issued by military organizations, was distributed to all the members of the party. This stated that the trucks would keep 100 ft. apart on the road, that they would not exceed a speed of 12

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m.p.h.; and that on hills the gears would be set for the speed change at which the trucks could surmount the obstacle.

The commissary equipment was borrowed from Squadron A, N. Y. National Guard, and was loaded on a light truck. It consisted of the National Guard field cooking equipment, tentage and all necessary supplies, together with two enlisted cooks.

The company after forming in the Borough of the Bronx proceeded west across the Borough of Manhattan to the Dyckman Street Ferry. After crossing the Hudson River, the trucks then climbed the Palisades to the state highways of New Jersey.

The first thirty miles or so of road were passed over in the dark entirely, night having fallen. It was one of the most interesting parts of the trip because the roads were unknown to any of the drivers. Without searchlights on the 5-ton trucks, they maintained

TRUCKS CONVOY DESCENDING PALISADES (FIRST TRIP)

a fair average rate of speed over a road covered with several inches of snow and succeeded in reaching the campground at New City, Rockland Co., N. Y., without any accidents.

We spent the night at the Squadron A farm, setting up our field kitchen out of doors, because there was no provision of any sort for cooking in the old farmhouse; on account of the extreme weather conditions we slept inside the farmhouse instead of pitching tents in the snow.

We did not refill our radiators the night before, nor did we make the adjustment of the machines at that time, as provided in the army regulations. It was late and dark when we arrived, and under those conditions, with hot engines and exhaust pipes, it was believed impracticable to make any adjustments. Such work should be done,

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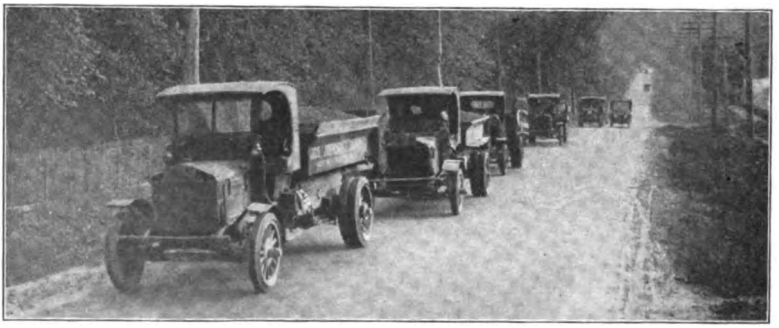
according to the opinion of those who took the trip, early in the morning and by daylight. As it was snowing and we did not know whether the temperature would go down below freezing, the radiators were emptied. One of the first things that the commissary corps did in the morning was to heat sufficient water to fill the radiators and make it possible to start the engines without difficulty.

Although the roads in southern New York and northern New Jersey are in the majority of instances improved, so-called, the center portion is usually improved and the ditches are treacherous. The commissary truck, which was sent about a mile to a garage where we had arranged to get our supply of gasoline for the return trip, became mired in coming back.

Two of the transport trucks standing on the improved portion of the road were connected by towing cables to the towing hooks on the front end of the commissary truck in the ditch, and by merely backing down the road a short distance the ditched truck was hauled



CONVOY IN CLOSE FORMATION ON SECOND TRIP

easily onto the road. We had provided a complete emergency equipment consisting of planks, timbers, jacks, towing cables, blocks and falls—everything that experience in operating fleets of motor trucks in and out of New York had indicated necessary.

On the return trip we passed over some bad roads on which we laid planks. In some places a tow-line from the leading truck, which was on firm ground, was used to pull the other trucks. This stretch of road was about a quarter of a mile long and it took over an hour to get the trucks through it. They were all driven through successfully and without damage.

At 11.30 a. m. on our return trip the commissary truck was given instructions to proceed until twelve o'clock, going as fast as the lighter load on the truck made possible, and then to stop and prepare the noon meal. This was done, and by the time that the convoy

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arrived everything was in readiness. Lunch was eaten and the commissary appliances and equipment repacked in the truck, a total delay of only 45 min. resulting.

The time of departure and arrival, the distance covered, the tonnage carried and the total number of ton miles for this trip are given in Table I. With the speed limited to 12 m. p. h., the average speed, including stops, both outward and on the return trip, was 5.3 m. p. h.; that includes all the time necessary to get through the difficult spots and to make all other stops.

The net average running speed, deducting the time taken for stops, was 6.2 m. p. h. It seems for country running a rather satisfactory average speed.

FARM LANE PASSED THROUGH ON SECOND TRIP

The moving factor, the ratio of running to standing time, was 86 per cent. This could be greatly improved as the result of experience. Experimental runs are advantageous because they give the owners of the trucks and all connected with the runs an idea how a company of trucks in convoy should be handled.

TABLE I—STATISTICAL SUMMARY OF FIRST TRIP

Time of departure 138th St., Mott Haven Canal	3.35 p. m., April 8, 1916
" " arrival at camp	9.40 "
" " departure from camp	10.08 a. m., April 9, 1916
" " arrival 138th St. and Mott Haven Canal	5.09 p. m.
Total time of trip	25 hr. 34 min.
" miles traveled	65.5
" tonnage (transport trucks)	20

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Ton-miles	1310
Time, out	6 hr. 5 min.
" in	6 " 16 "
" in camp	12 " 28 "
" noon mess	— " 45 "
Net running time	10 " 36 "
" standing time en route	1 " 45 "
Average speed	5.3 m.p.h.
" " out	5.3 "
" " in	5.3 "
Net average running speed	6.2 "
" " " " out	6 "
" " " " in	6.4 "
Moving factor (ratio running to standing time)	86 per cent
Maximum non-stop run	8.5 miles
" running speed (for 35 min.)	12.4 m.p.h.

DESCRIPTION OF SECOND TRIP

The second trip was made by another set of owners, who are in the contracting business. This run was made in Westchester County, New York, and in Connecticut. The roads were good most of the way. The destination was a point directly north of the Kensico Reservoir, which is part of the reservoir system of the New York water supply. This point would necessarily have to be defended against any attack on New York because the interruption of the water supply of a besieged city is one of the best means of capturing it. In time of attack on New York all these contractors' trucks might be running to points similar to this, carrying structural steel, cement, and various building materials to be used for defenses, fortifications, and emplacements for guns; it is appropriate therefore that this trip should have been made by owners of trucks suitable for the transportation of such material.

On both of these runs we were unfortunate or perhaps fortunate in having adverse weather conditions. On the second run, which was made at the end of May, a heavy downpour of rain during the night rendered the roads extremely slippery. Going into the campground we left the main road, passed over a typical farm road and through the farmyard. We encountered some difficulties in getting back through that farmyard owing to the rainfall of the night.

The emergency truck on which towing appliances were carried met with misfortune just as it was leaving the farmyard; the rain on the sloping road permitted the truck to slide into the ditch, and some little difficulty and delay was encountered in getting it out.

On the first run, a truck in difficulties was always assisted by one

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of the other trucks. But the man commanding the second expedition believed there was a quicker method; he used a little runabout that belonged to one of the orderlies of the company. The runabout had a cable hooked onto it; one of the heavy trucks, on solid ground, was used as an anchor, and the other truck, which was mired nearly to its hubs, was pulled out by the runabout.

The weight of these trucks had no adverse effect upon their tractive ability and was not the cause of their being mired. In most instances the narrow front tire was the one that sank first into the ground. The wide rear tire seemed to roll much better over the bad places.

Noon mess occupied about the same time as it did on the previous trip. Some of the drivers discovered then that they needed gasoline and consequently there was a considerable wait while the tanks of

RUNABOUT HAULING OUT MIRED TRUCK (SECOND TRIP)

gasoline, which had been carried on the emergency truck, were tapped and the various trucks supplied.

On the way back the drivers drove the trucks as fast as the engine governors would permit; one new truck, which could make about 18 or 20 m. p. h., was always in the lead, while a truck about four years old, which was not in especially good condition, was always in the rear. It seems that trucks, even of the same make, can hardly be operated successfully in convoy, unless they are in the same condition and their governors set for the same speed.

A summary of the second run is given in Table II. The moving factor on this trip was 56.7 per cent, very low indeed. The arrange-

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ments for the first run were perhaps a little superior to those for the second. The fact that the trucks had to pass through a farmyard was the cause of our only serious delays on the second trip.

STATISTICAL SUMMARY OF SECOND TRIP

Time of departure 177th St. and Harlem River, Morris Heights	3.20 p. m., May 24, 1916
" " arrival at camp	7.30 "
" " departure from camp	11.37 a. m., May 25, 1916
" " arrival at Morris Heights	3.50 p. m.
Total time of trip	24 hr. 30 min.
" miles traveled	54.5
" tonnage (transport trucks)	26.4
Ton-miles	1446
Time, out	4 hr. 10 min.
" in	4 " 13 "
" in camp	12 " 35 "
" forming company for return	3 " 27 "
" noon mess	— " 35 "
" filling gasoline tanks	— " 45 "
Net running time	5 " 21 "
" standing time en route	3 " 2 "
Average speed	6.5 m.p.h.
" " out	6.4 "
" " in	6.6 "
Net average running speed	10.2 "
" " " " out	9.5 "
" " " " in	11.1 "
Moving factor (ratio running to standing time)	56.7 per cent
Maximum non-stop run	13.6 miles
" running speed (for 1 hr.)	15.9 m.p.h.

MILITARY AUTHORITIES APPROVE EXPERIMENTS

Not only have General Wood, Commander of the Department of the East, and General O'Ryan, Commander of the National Guard Division of New York, given their approval and cooperation in these experiments, but the Hon. Newton D. Baker, Secretary of War, has given an interview to a committee of the motor-truck owners, and has invited the cooperation of a committee of these owners with a committee of the General Staff of the United States Army.

It is suggested that similar experiments and test runs can with advantage be carried on in localities where there are a sufficient number of motor-truck owners who are interested in this practical form of preparedness, and which localities may at some future time be within a theater of military operations in the event of war.

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THE FARM TRACTOR

By C. M. Eason
(Member of the Society)

Abstract

Inasmuch as horses cannot meet the demand for increased farm power, the tractor must come at once. So far the supply of tractors has been entirely inadequate to meet the demand. The author specifiies some of the problems that confront designers of farm tractors. To make the tractor immediately available for farm work, it must be adaptable to practically all of the existing types of horse-drawn implements, besides furnishing belt power for a wide variety of present power-driven farm machinery. In designing tractors it must be remembered that the horse is a very flexible unit, capable of a wide variation in power output.

Designing a tractor to furnish the necessary power for the majority of farm conditions, requires an intimate knowledge of crops, soils and farm management. These must be analyzed carefully so as to make the machine have as wide a range of usefulness as possible. The power required for operations in connection with various crops, is outlined, as are also the conditions that must be met in plowing and cultivating corn, wheat, oats and other crops. The power required for pulling plows varies from 250 to 1500 lb. per bottom, depending upon the condition and kind of soil, depth and type of plow.

The author reviews what the tractor must show in the way of economical operation, pointing out that for permanent success it must show higher returns on the investment than horses. Information is also given in regard to the mechanical efficiency, engine efficiency, friction losses, rolling resistance and general design requirements. Other factors peculiar to farm tractor work, which must be taken into account, are treated. In conclusion photographs and a brief description are given of representative tractors.

LAND FOR FOOD SUPPLY

The total arable land area of the United States is in round figures two billion acres. At the present time about nine hundred million acres are devoted to farming, of which only five hundred million acres, or one-fourth of the total, are classified as improved farm land. Considering the tillable land available for cultivation, in addition to the unimproved farm lands, less than one-half the possible farm resources of the country has been developed. The remaining resources must be made productive, and the available supply of foodstuffs thereby materially increased if the rapid development of the nation as a whole is to be continued.

Within the past few months, due to the war in Europe the necessity of increasing our food supply has become apparent. The countries affected by war have to-day practically no reserves of food.

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The exportation of food and of horses to Europe has recently risen to considerable proportions. The indications are that these exportations will be increased, rather than diminished, after the war is over.

PRESENT FARM POWER

Farm production, or more specifically the raising of foodstuffs, is largely a question of power. Aside from windmills used for

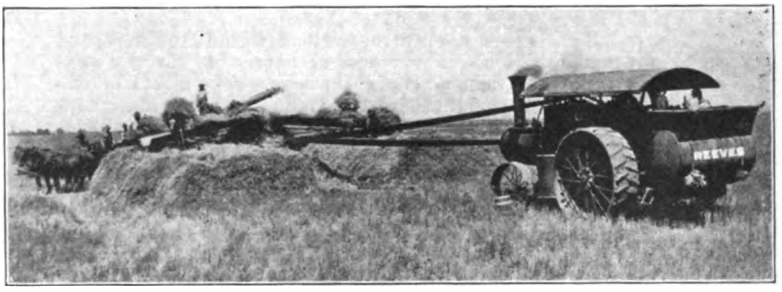

FIG. 1—STEAM TRACTOR AT WORK THRESHING

pumping water the available farm power equipment of the United States consists of about twenty-five million horses and mules, sixty thousand gas tractors, a hundred thousand steam tractors and one million stationary gasoline engines. This represents a total of over thirty million horsepower, which is more than the combined horsepower of the powerplants of all the other industries. Large as this total is, however, it allows only one horsepower for each 30 acres

FIG. 2—EARLY TYPE OF STEAM TRACTOR-PLOWING

of farm land, which is not enough power by perhaps one-half or one-third.

Aside from the additional power required to put more land under cultivation, it has been demonstrated scientifically that the production of the present acreages could be increased by deeper plowing, better preparation of the seed bed, more timely planting and more intensive cultivation, all of which call for a greater expenditure of power.

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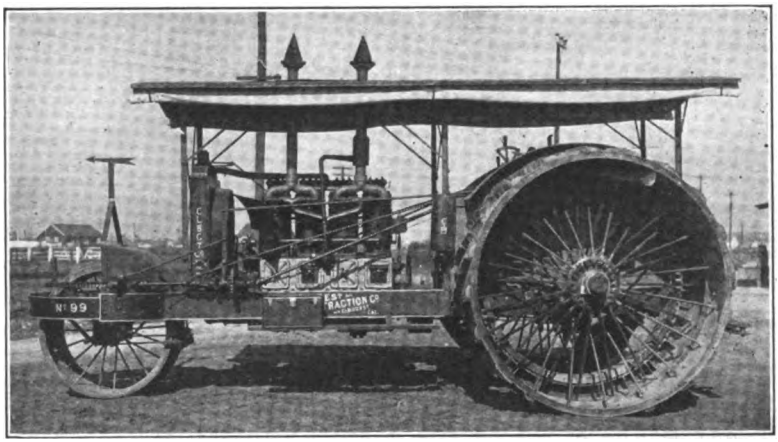

FIG. 3—C. L. BEST ROUND-WHEEL TRACTOR

It has been estimated that the improved farm lands are now producing only 40 per cent of their potential capacity, and that by having sufficient power available their production can be increased to at least 60 per cent of the potential capacity.

HORSES CANNOT MEET DEMAND

The horse is now the chief source of power for farm work. It takes nearly five years to produce a horse capable of doing heavy farm work, and to provide enough horses to meet the immediate demand is impossible. The war has withdrawn over one million horses from the United States in the past twelve months and at

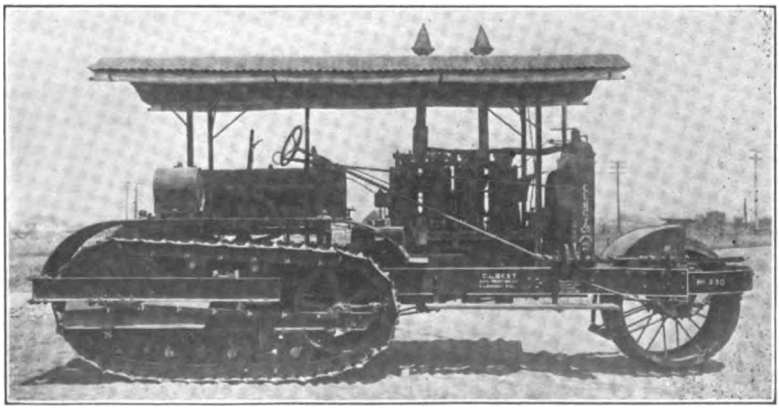

FIG. 4—C. L. BEST ENDLESS-TRACK TRACTOR

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the present rate of increase it will take six years to replace these alone. The horse has other drawbacks, principal of which is the fact that one-fifth of all the farm output is required to feed him. Each horse takes the product of about five acres of ground for his

FIG. 5—AULTMAN-TAYLOR CO., 30-60 GAS TRACTOR

maintenance. The demand for horses has been so great that their cost has increased 143 per cent in the past ten years. If the enormous development of the automobile and the motor truck had not relieved the situation in a large measure, the price of horseflesh would be still higher and the shortage of farm power felt more keenly.

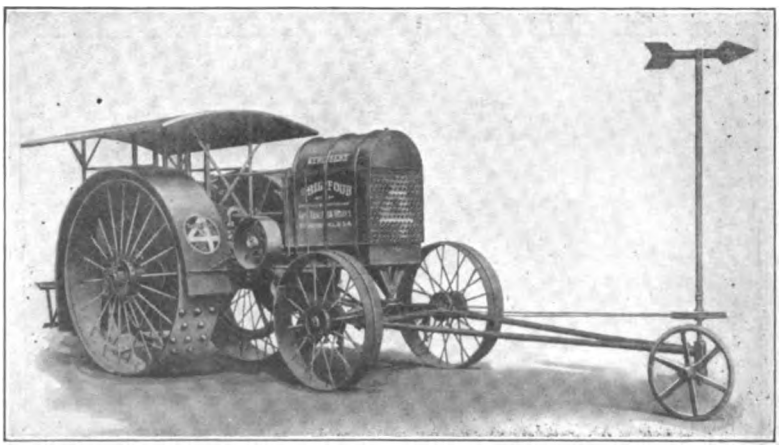

FIG. 6—BIG 4-"30" GAS TRACTOR—EMERSON-BRANTINGHAM CO.

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Horses have already been replaced by mechanical power almost exclusively for certain classes of work, such as heavy threshing, which is done largely by the steam traction-engines or the large gasoline tractors. The small stationary gasoline engine has replaced the horse entirely for such odd jobs about the farm as sawing wood, grinding feed and pumping water. The heavy threshing engines and the small stationary gas engine meet farm power requirements for which the horse was always fundamentally unsuited. To obtain permanent relief from the farm power situation, machines capable of doing the work now done by horses are absolute necessities. Between thirty and forty thousand gasoline tractors with an average

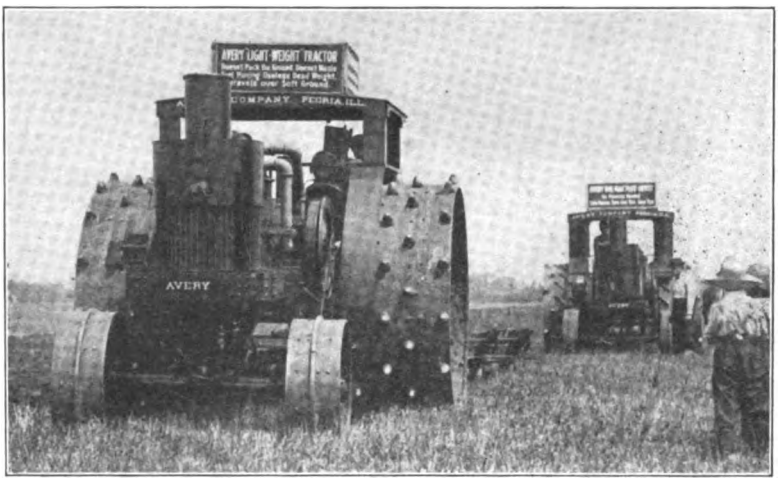

FIG. 7—PAIR OF AVERY 30-60 GAS TRACTORS

drawbar pull equivalent to that of six or eight horses have been produced during the past few months. The supply of these tractors has been entirely inadequate to meet the demand and the indications are that more than twice the number mentioned will be built next year.

PROBLEMS OF FARM TRACTOR DESIGN

To design a machine capable of replacing animal power for farm work is an extremely complex problem. To make it immediately available, such a machine must be adaptable to practically all the existing types of horse-drawn implements, besides being able to furnish belt power for a wide variety of present power-driven machinery used on the farm. The amount of power required to operate both horse-drawn implements and belt-driven farm machinery is extremely

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variable. This is because comparatively little has been done toward standardizing constructions and because of the wide differences in soil and crop conditions, which affect the draft of all implements and the power load of nearly all belt-driven machines. The horse is a flexible power unit and these differences in power requirement have not been particularly apparent when using horses. Three-quarters of a horsepower is the average output of a horse, but for short distances he can deliver five or six horsepower, and often in the rush season of farm work will deliver one and one-quarter to one and one-half horsepower for several days. He also has a considerable range of speed as well, and possesses a nimbleness of action

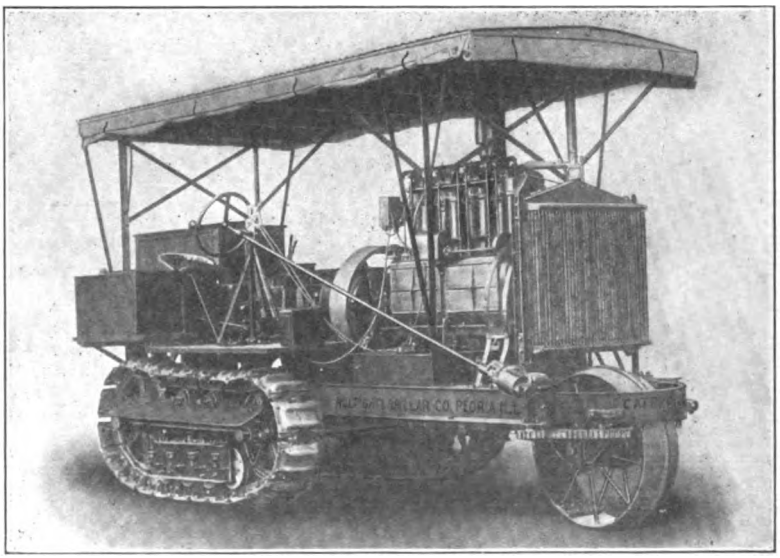

FIG. 8—HOLT-CATERPILLAR 75-HP. TRACTOR

difficult of attainment with a machine. Having a certain amount of intelligence a horse is readily trained to follow a furrow or a crop row, thereby relieving the operator of considerable work otherwise necessary to guide a machine properly.

Since practically all existing farm implements have been developed to take advantage of animal power, it is essential that the tractor be designed so that its power is available in the form of horizontal draft. Much of the machinery now used with horses is fundamentally unsuited to the efficient use of mechanical power. There are vast possibilities of development of new types of agricultural machines to take advantage of the direct application of mechanical power.

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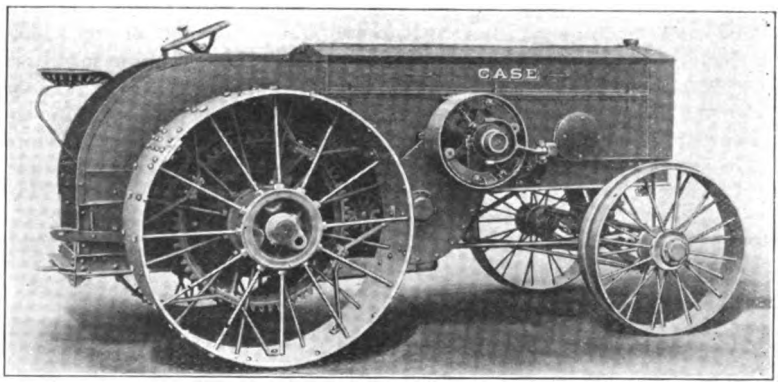

FIG. 9—"20-40" TRACTOR, J. I. CASE T. M. CO.

These are however problems for the future. The immediate necessity is to provide a machine that will operate as many of the present horse-drawn implements as possible and thereby make available immediately greater power resources for increasing the available food supply. The variety of horse-drawn implements is almost endless. One large implement manufacturer finds it necessary to make and stock over one thousand different models of plow to meet the varying conditions of soils and farm methods. The plow is only one of several hundred different kinds of agricultural implement, each having

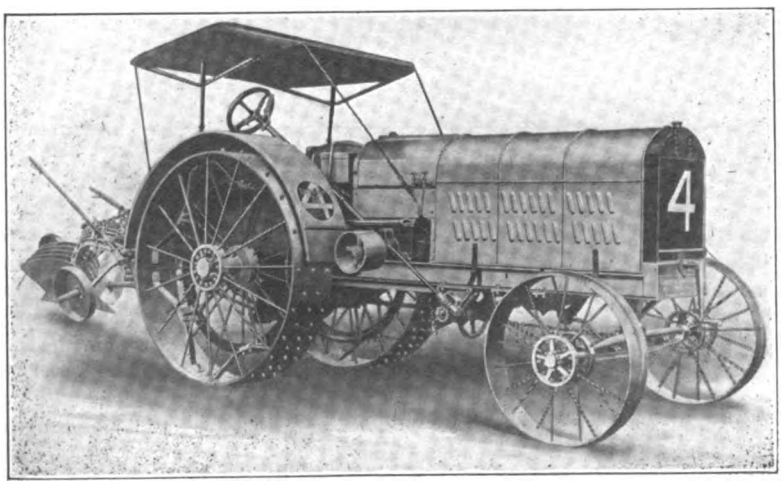

FIG. 10—"BIG 4-20" TRACTOR, EMERSON-BRANTINGHAM CO.

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its own peculiar range of usefulness. Every crop raised seems to need some machine peculiar to it alone, and on account of the widely differing soil and moisture conditions, peculiar to certain localities, each crop seems to require several different kinds of machine. To design the ultimately satisfactory tractor that will furnish power for the majority of farm conditions throughout the country requires an intimate knowledge of crops, soils and farm management. It will be nec-

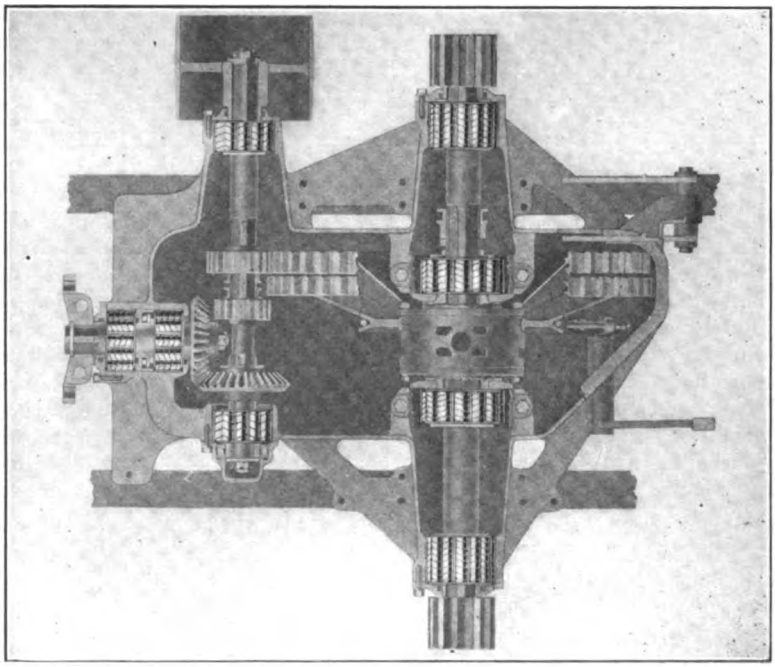

FIG. 11.—TRANSMISSION EMERSON-BRANTINGHAM CO. TRACTOR—PLAN VIEW

essary to analyze carefully the work that must be done on a farm from beginning to end of the crop cycle. It will be necessary also to make a careful study of the various implements used, in order to provide in the tractor sufficient flexibility of operation and of methods of application. Adaptability, or range of usefulness, is one of the most important factors in farm tractor design.

In the United States as a whole a little more than one-half of the acreages devoted to cereals is in corn, a little less than one-fourth in wheat, and slightly less than one-sixth in oats. Table I shows the acreage and values of the principal crops produced in 1910 as given by the census of that year. Corn is therefore the leading crop.

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POWER REQUIRED FOR VARIOUS CROPS

To raise corn successfully requires the application of more work per acre, measured in horsepower-hours, than does the raising of any other crop. The methods used vary greatly; they are briefly: First—Preparation of the seed bed by plowing, pulverizing and dragging, or rolling. Second—Planting. In the corn-belt States east of the Mississippi River most of the corn is planted with check-row planters. Between the Mississippi and Missouri Rivers both the check-row system and planting by the corn lister are followed; while

TABLE I—VALUE AND ACREAGES OF PRINCIPAL CROPS PRODUCED IN 1910

Crop	Acreage	Value
Corn	105,825,000	$1,565,258,000
Wheat	49,543,000	543,063,000
Hay	43,017,000	694,570,000
Oats	37,763,000	414,663,000
Cotton	36,681,000	800,000,000
Barley	7,627,000	139,182,000
Potatoes	3,619,000	833,778,000
Rye	2,127,000	87,557,000
Tobacco	1,012,800	85,210,357
Buckwheat	833,000	12,735,000
Rice	693,300	18,224,000

west of the Missouri River, in the Nebraska and Kansas corn belt particularly, nearly all of the corn is listed. The corn rows are between 36 and 44 in. apart. In the extreme South an acceptable method is to plant the corn in single drill rows about 5 ft. apart. When the corn begins to come up it is thinned out by hand, leaving only the best stalks standing from 6 to 12 in. apart. Third—Cultivation. Necessarily corn cultivation is modified by the type of planting.

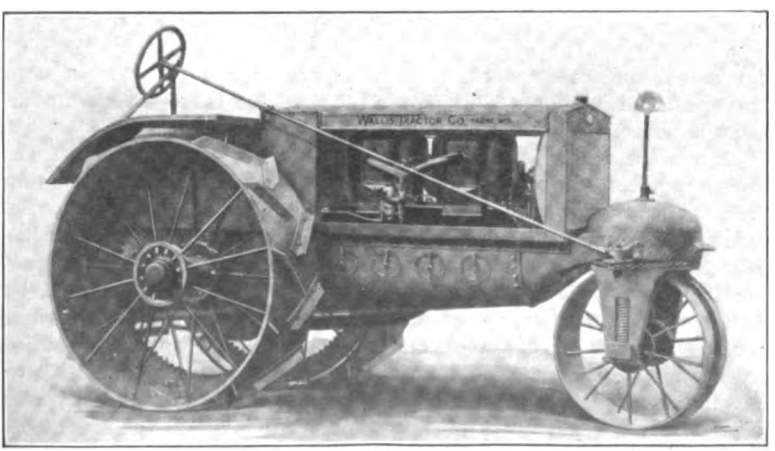

FIG. 12—WALLIS "CUB" TRACTOR—MOUNTED ON SPRINGS

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The number of times the field is gone over with the cultivator depends on the soil and moisture conditions, being generally less in the Central and Eastern sections, and more in the Western and more arid localities. There are hundreds of different corn cultivators, different soil conditions seemingly requiring altogether different shovel construction. In the West a cultivator having disks instead of shovels is used to a considerable extent. Fourth—Gathering or Harvesting. Where corn is raised for grain most of the gathering is done by hand, either from the field of standing stalks or from the shock, the corn having been cut either by hand or by the corn binder. Fifth—Preparation for Feeding. This varies greatly, depending upon the kind of feeding practice. Silage feeding is becoming popular and corn is used extensively for this purpose, the stalks and grain being cut into ensilage. Where the grain itself is fed whole to cattle or ground into feed

Fig. 13—Front View of Wallis Cub Tractor

for horses and other stock, it is necessary to run the ears of corn through a corn sheller. The power required for these various operations is shown in Table II.

It can be seen from the foregoing incomplete analysis that the power requirements for raising corn are extremely variable, ranging from a heavy power-plowing outfit to a one-horse cultivator. Because of these discrepancies in size of power units, the prediction has been made that the horse could never be replaced for corn raising, particularly in view of the fact that the horse guides himself by following the corn rows when pulling a cultivator. The most serious problem of applying power to corn cultivation is the matter of control or of guiding the machine, since the operator of a corn cultivator has to be occupied chiefly with the proper operation of the cultivator shovels. It is difficult to steer a tractor and watch the cultivator shovels at the same time. This difficulty might possibly be overcome

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by providing two men, one to drive the tractor and the other to operate the cultivator. By using a double-row cultivator and taking into account the increased speed and more continuous operation of the tractor, it is probable that two men could get more work done in a day than if each were running a horse-drawn cultivator.

For wheat, oats and other small grains the various operations are more nearly uniform as to power requirements. The principal

TABLE II—POWER REQUIRED FOR RAISING CORN

Operation	Implements and Power Used
Preparation of Seed Bed:	
Plowing	Three horses on one 14-in. plow.
	Five horses on two-bottom gang plows.
	Tractors of various sizes on plows ranging from two to ten bottoms.
Disking and harrowing	Three to four horses on single 6-ft. disk harrow.
	Five to seven horses on double 6-ft. disk harrow.
	Light tractors only. Harrowing usually done at same time as plowing.
Planting:	
Check-row planting	Two horses only.
Listing	Three to five horses, depending on soil condition.
	Six to eight horses on double-row lister.
Cultivation:	
Hoeing	Hand work, not much used.
Scuffeling	One horse on five-hoe cultivators. Works between rows.
One row cultivation	Two horses.
Double row cultivation	Three horses.
Lister cultivation	Four horses.
Gathering:	
Hand husking	Team, wagon and one or two men.
Corn binder	Three to five horses and three men.
Feed Preparation:	
Corn shelling	Hand power, belt-driven sheller requiring 5 to 10-hp. engine.
Ensilage cutting	Cutter requiring from 8 to 40 hp.
	Horses, wagons and men for hauling.

wheat and small grain belt lies between the Rocky Mountains and the Mississippi River; the Dakotas, Kansas and Oklahoma being the leading wheat centers. Owing to soil and climatic conditions and to the fact that these States are comparatively level, these crops are usually found in large fields, and it is therefore necessary that the power equipment be extensive. Table III gives an incomplete analysis of the power requirements and methods used for raising wheat and other small grains.

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Tables II and III indicate the magnitude of the problem of applying mechanical power to farming operations. They represent only two kinds of farming—there are many other kinds of combinations, the power needs of which must be met. It may be impossible to combine in one type of tractor all of the essential elements necessary to meet the entire range, but the ideal tractor would meet practically all farm power requirements. It seems likely however that several general types of tractors will be necessary.

DRAFT OF FARM IMPLEMENTS

The power required for pulling plows varies from 250 to 1500 lb. per bottom, depending on the kind and condition of the soil, depths, and the type of plow used. The average 14-in. mold-board plow, working 6 in. deep, requires three horses. The average draft of a

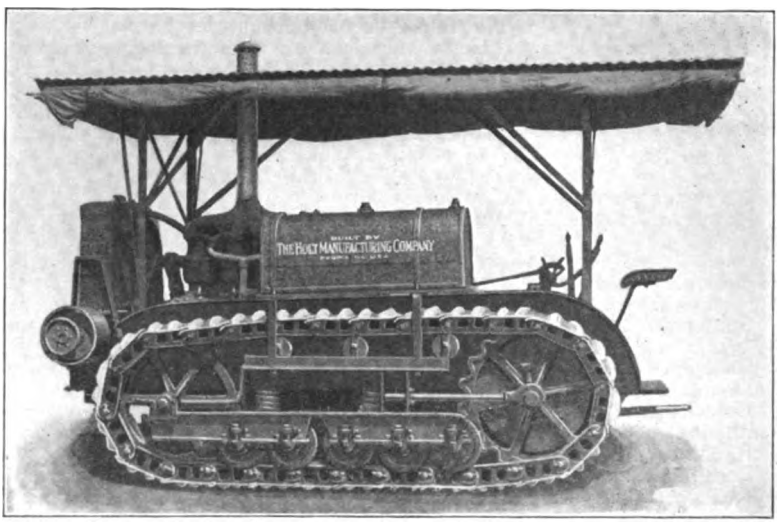

FIG. 14—HOLT 45-HP. CATERPILLAR TYPE OF TRACTOR

horse is 150 lb., and his walking speed about 1.8 miles per hour. On this basis it is usually assumed that a 14-in. plow requires a pull of only about 450 lb., but this is practically always exceeded on account of the fact that in plowing, horses are worked considerably beyond their average draft. Actual dynamometer tests of three horses pulling a single 14-in. plow show a drawbar pull as high as 600 lb., or 200 lb. per horse. In considering the problem of applying tractor power to plows, it is not safe to figure on much less than 700 lb. average drawbar pull per plow, and it is necessary to provide sufficient power in the tractor to exert a pull of 1000 lb. per bottom for average conditions of plowing. This is due to the fact that in a field where

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the average pull per bottom will be 650 or 700 lb., hard spots or tough places will often increase the draft to from 950 to 1000 lb.

Little information is available in regard to the power required to pull other farm implements and the field for experimental research work in this direction is immense. Plow and implement manufacturers are studying this problem at the present time and are constantly working toward higher efficiency in their machines.

The future will undoubtedly see many of the existing farm tools modified to make them more adaptable to the tractor, and capable of doing more work with a less expenditure of power.

TABLE III—POWER REQUIRED FOR WHEAT AND SMALL GRAIN

Operation	Implements and Power Used
Preparation of Seed Bed:	
Plowing	Two to six horses used on one and two-bottom plows.
	Steam and gasoline tractors used on all sorts of plows from two to twelve-bottom. Deep tilling disk-plows sometimes used.
Disking	Three to seven horses on 6 and 8-ft. single and double-disk harrows.
Planting:	
Drilling	Two to four horses on grain drill.
Harvesting:	
Binder	One man and four or five horses on 6 to 8-ft. binder.
	Tractor pulling one to five binders, one man on each binder.
	Sometimes two horses to pull binder and 5-hp. gasoline engine for running binder mechanism.
Header	Five to seven horses on 10 to 16-ft. header. About five teams and wagons, each with two men to haul and stack grain.
Combined harvester and thresher	Eighteen to thirty horses or mules.
	Special tractors, both steam and gasoline, on West Coast exclusively.
Threshing:	
Grain separator	Belt-driven, practically always by stationary, portable and tractor engines of all kinds. From 10 to 120 hp.
Disposal:	
Hauling to market	Horses and wagons used almost exclusively. Place here for road tractors, trucks and tractor trains.

ECONOMY OF OPERATION

To be a permanent success tractors must show higher returns on the investment than horses. The conclusions of the United States Department of Agriculture seem to be that if the tractor is to be an economical investment for the farmer, it should cost about one-half as much as the horses it replaces. This is based on a 20-per cent depreciation annually for the tractor as against 7.5 to 10 per

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cent for the horse. The investigation on which these conclusions were based covered tractors that had been actually used two or more seasons. Unfortunately the only tractors available for study at the time of the investigation were some of the earlier designs, which were crudely constructed, inefficient in the use of fuel and operated for the most part by untrained drivers. As tractors improve in design, efficiency and reliability the depreciation mentioned will no doubt be decreased and a higher first cost per horsepower be made possible.

The first cost of a tractor will be established eventually by the amount of work the tractor is capable of doing. One of the principal

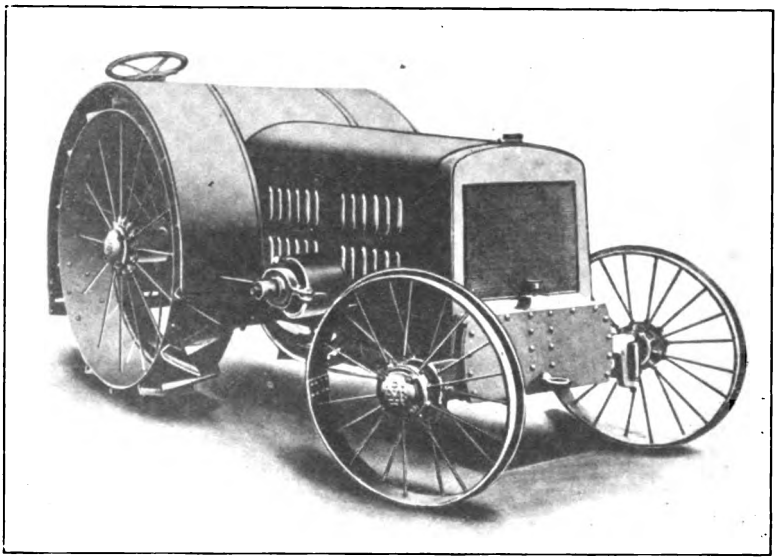

FIG. 15— LARGER SIZE OF MAYER LITTLE GIANT TRACTOR

reasons for changing from horse to tractor methods of farming is that the tractor enables one man to do more work in a given length of time. One man and five horses with a two-bottom plow, which is about the largest practical unit, can plow only an average of four to five acres per day. One man with a tractor pulling three or four plows can, because of the higher average speed of the tractor, plow ten to twelve acres per day. On the basis of a 20-per cent depreciation for the tractor against 7.5 per cent for the horses, the work value of the tractor is practically equal to that of the horses, and has the additional value of reducing the man labor by one-half. The lowest average cost of five work horses is about $750 and it therefore seems that a tractor capable of doing twice as much work per day as the horses would be worth equally as much.

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In addition to first cost there is also the problem of operating cost, both as regards fuel consumption and repairs. Government statistics indicate that the cost of keeping a horse is about $100 per year. Little authoritative information regarding tractor operating cost is available, but it is generally believed that for an equal amount of work the tractor costs somewhat less than horses. It is highly improbable that the present cost of horse farming can be reduced; in fact the indications are that it will increase. The cost of tractor operation will be reduced by increasing the mechanical efficiency and reducing the repair cost by a general refinement of the design.

MECHANICAL EFFICIENCY

Since a tractor must be considered as a machine for converting heat units into horizontal draft, the power losses can be classified

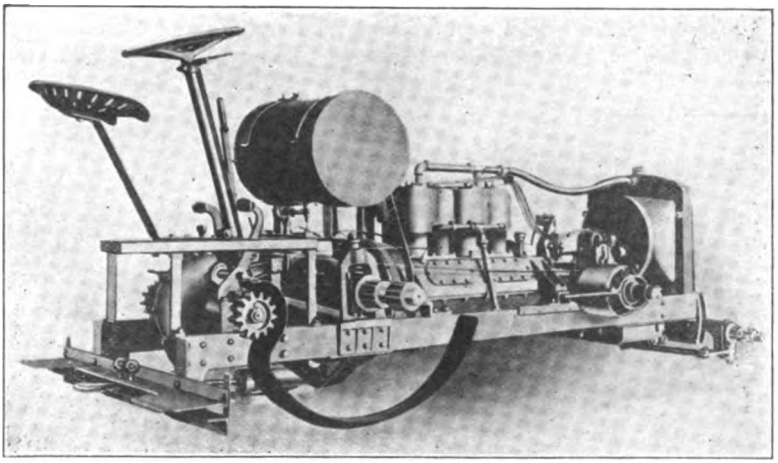

FIG. 16—POWERPLANT OF MAYER LITTLE GIANT TRACTORS

as follows: Thermal and mechanical losses of the engine; friction losses in the transmission system; and rolling resistance due to propelling the tractor.

In considering these factors I assume that the farm tractor will consist primarily of an internal-combustion engine for producing power from the heat of the fuel; a transmission system for delivering the power of the engine and reducing its speed to that of the driving wheel or other tractor mechanism; and means for propelling the tractor and pulling other implements.

The net mechanical efficiency of a tractor is of course the percentage of power developed by the engine and delivered at the drawbar. The drawbar pull of a tractor is the product of the engine

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torque, gear ratio and transmission efficiency, divided by the radius of the drive wheel, and less the rolling resistance, which is the product of the total weight times the percentage of resistance. Reduced to a formula this is as follows:

$$Dr = \frac{T \times E \times R}{D} - (W \times Tr)$$

Dr = Drawbar pull in pounds.
T = Torque of engine in pound-inches.
E = Efficiency of transmission system.
R = Ratio of gear reductions, engine to drive wheel.
D = Driving wheel radius in inches.
W = Weight of tractor in pounds.
Tr = Percentage of tractive resistance.

ENGINE EFFICIENCY

To obtain the lowest fuel consumption, engines for tractors must be designed to give the highest possible efficiency. In order to keep

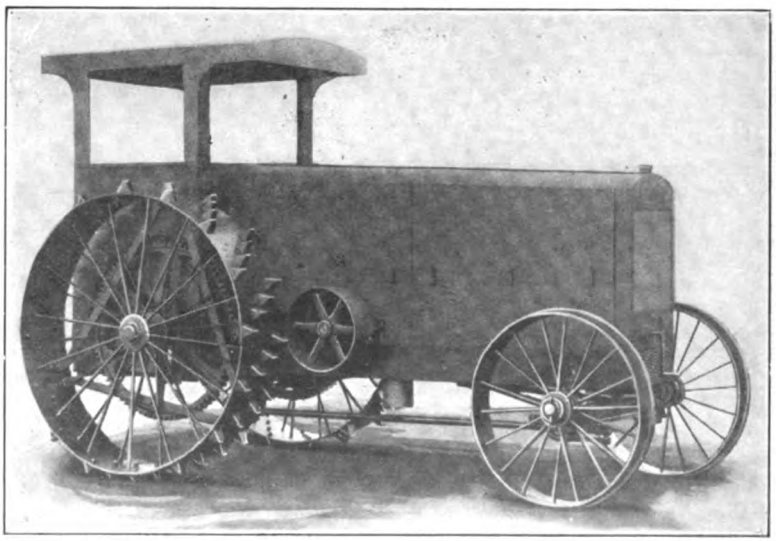

FIG. 17—LAUSON 15-25 TRACTOR OF THREE-PLOW CAPACITY

the total weight as low as possible, tractor engines must have a high volumetric efficiency, as well as low fuel consumption. It is essential that the average power output be sufficiently under the maximum to insure an adequate peak-load capacity.

The working load of a tractor engine used for plowing varies 25 to 35 per cent. It is impracticable to provide for the variation by

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changing the speed ratio in the transmission. The reserve capacity must therefore be provided in the engine itself. A commonly accepted method of establishing the size of engine is to design the tractor so that the normal working load will be about two-thirds of the maximum engine capacity. The tractor should travel at practically a constant speed, which necessitates that the engine be governor-controlled. With a fixed speed and a reasonably constant load of two-thirds the maximum engine capacity, it should be a comparatively easy matter to design the valve-timing, valve sizes and manifold, so as to obtain maximum thermal efficiency.

FRICTION LOSSES

The efficiency of the tractor transmission system varies with the type of construction. In some of the earlier tractors rough cast gears

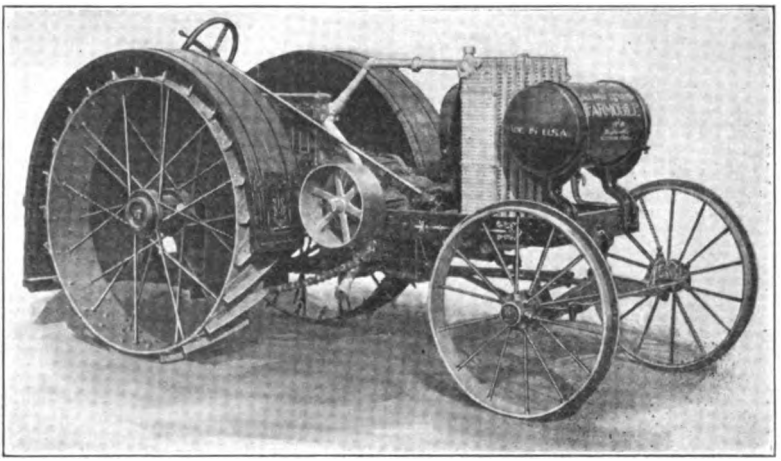

FIG. 18—WM. GALLOWAY FARMOBILE—TRANSMISSION IN FRONT OF ENGINE

of large diameter, coarse pitch and open to dirt and dust were used. As they were mounted on shafts carried in plain bearings that were subject to mis-alignment and had inadequate provision for lubrication, the friction loss amounted to about 10 per cent of the engine power for each pair of gears in mesh.

In the type of transmission now used in tractors of modern design cut and hardened gears run in dust-tight oil-proof cases. All journals are carried in high-grade anti-friction bearings. In this transmission, which is common to automobile and truck construction, the friction loss is not over 2 or 3 per cent for each pair of gears. Further increase of efficiency in this transmission system is largely a matter of keeping gear sizes down to reasonable proportions and limiting the number of gears in mesh.

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A working speed of 2.5 miles per hour, with a drive wheel 4.5 ft. in diameter, requires a gear ratio of approximately sixty-five to one, when the engine runs in the neighborhood of 1000 r.p.m. A common method at the present time is to use three sets of gears in the transmission system, making the first two reductions with cut and hardened gears running in a self-contained case and the final reduction with cut and hardened pinions running in cast bull gears of large diameter, mounted directly on the rear wheels. In many of the present designs these final reductions are left exposed, but in some of the latest tractors they also are enclosed.

ROLLING RESISTANCE

The power necessary to overcome the rolling resistance varies directly as the total weight when the conditions of soil surface and

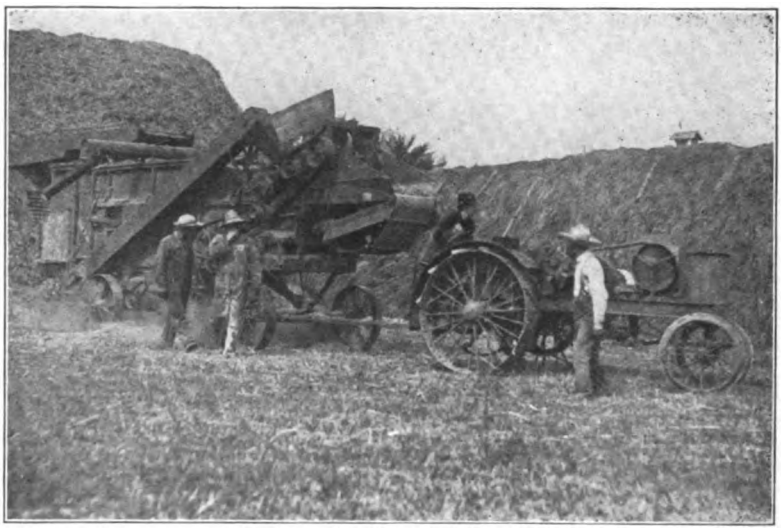

FIG. 19—WATERLOO BOY TRACTOR AS USED FOR STATIONARY WORK

the pressure per unit of ground contact remain constant. Under varying conditions of soil surface, which allow the wheel to sink to different depths, the rolling resistance increases in proportion to the depth to which the wheel sinks. The rolling resistance is inversely proportional to the ground contact, being less for large-diameter and wide-face wheels than for small-diameter narrow-face wheels under average conditions. In soft deep mud the greater ground contact results in greater rolling resistance, but for all average soil conditions the greater area is of advantage since it permits of a lower unit pressure and thereby prevents sinking of the wheels.

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Since there is a practical limit to the diameter and the width of face of wheels, a number of endless tracks, or caterpillar treads, have been designed. The object is to reduce the unit pressure of ground contact. Although strong claims are made for the increased efficiency of this system, its use has not become general because of patent restrictions.

The power required to propel the tractor is increased in direct proportion to the grade, being about 1 per cent of the total weight for each 1 per cent of rise, and this grade resistance must be added to the rolling resistance. It will be seen that in order to get the greatest possible efficiency by reducing the rolling resistance it is

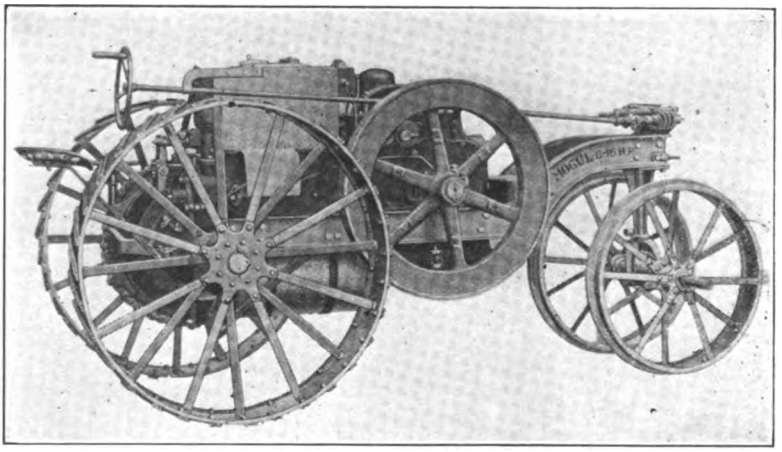

FIG. 20—INTERNATIONAL HARVESTER CO., 8-16 MOGUL

necessary to keep the total weight as low as possible and have the area of ground contact as great as practical. Reduction of total weight can only be carried to the point where the power transmitted equals the holding power of the wheels. The pulling ability of a tractor depends entirely on the number of square inches of hold that can be obtained on the ground by the use of spurs, grousers or mudlugs on the face of the wheel. It also depends on the holding power or tenacity of the soil surface. The spurs or lugs which are necessary on the driving wheels to secure traction, cause additional rolling resistance, adding to the power required to propel the tractor. In order to be effective they should be applied, or formed, so as to enter and leave the ground as nearly perpendicularly as possible. A common form of spur, or cone, has an involute surface. Lugs made of angle iron shaped to fit across the face of the wheel at an angle of about 45 deg. and bent slightly forward so that they leave the

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ground in a perpendicular direction have been found to be more efficient in very soft ground than any other type.

As far as can be ascertained the average rolling resistance for tractors having ordinary wheels and conventional lugs is about 15 per cent of the total weight in stubble plowing and about 25 per cent on recently plowed or very soft ground. The allowable pressure on the face of the wheel is in the neighborhood of 25 lb. per sq. in. of ground contact on stubble ground and probably not over half this amount on recently plowed ground.

To prevent excessive wheel slippage the pressure on the face of the grousers should not be over 75 lb. per sq. in. for average soil conditions. These figures are only approximate since not many data

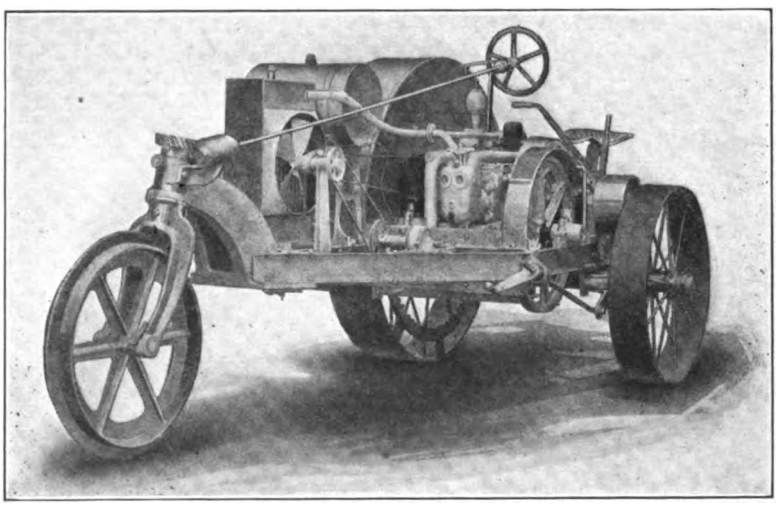

FIG. 21—BULL TRACTOR WITH SINGLE DRIVING WHEEL

are to be had on the subject and a great deal is yet to be learned. Reducing the rolling resistance is a study in itself requiring a great amount of experimental research work before definite values can be fixed.

GENERAL REQUIREMENTS

A tractor must be so designed that it will operate an entire season without adjustment or replacement of any of the wearing parts. This means an average of 1000 hours of service with an absolute minimum of attention. This necessitates exceptionally large bearing areas for the main bearings and connecting-rods of the engine and a lubrication system of simple design and practically infallible operation.

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A tractor engine must operate at nearly full power while traveling so slowly that there is little or no natural draft through the radiator. The cooling system must be designed to meet these conditions. The radiator itself should be as nearly "clog-proof" as possible, since it is often difficult to secure clean water, and many operators are indifferent as to the amount of foreign matter introduced into the cooling system with the water. It is obvious that the radiator must be ruggedly built to withstand the rough service to which it is subjected.

The carbureter of a tractor engine must be as simple as possible, and designed so that frequent adjustment is unnecessary to give the highest possible fuel economy. The matter of burning kerosene in tractor engines is worthy of serious attention. If the present high price of gasoline continues it will be necessary for tractor engines to be equipped for burning kerosene.

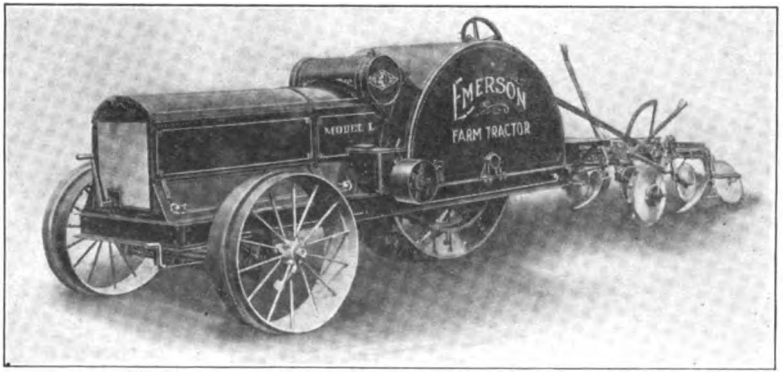

FIG. 22—EMERSON MODEL "L" TRACTOR WITH SINGLE DRIVE-WHEEL

Since the tractor must operate in a continuous cloud of dust it is essential that the air going through the carbureter into the engine be cleaned and some sort of dust-separator provided. The problem of protecting all of the working parts against this dust naturally affects the design as a whole. To insure long life with a minimum of replacements, dust should be guarded against at every possible point in the tractor construction.

The pulling of plows and other farm implements subjects a tractor to severe vibration, to overcome the destructive effect of which the tractor as a whole must be constructed of the fewest possible parts. These parts must be held together by bolts of sufficient size to overcome any possibility of their working loose.

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TENDENCIES IN DESIGN

These are only a few of the problems involved in the design of successful farm tractors. The big problem of tractor design is the one of establishing a type, or the types, that will meet the requirements of farm operation over a wide range of soil and crop conditions. The secondary problem is to produce this tractor at a sufficiently low selling price to make its use comparatively economical. The development and refinement of the mechanical details of tractor construction are relatively simple. The technical information necessary to design an efficient power unit has already been provided by automobile engineers, by motor-truck engineers particularly. The great need of the tractor industry at this time is to combine this technical information with the information already secured through practical experience in

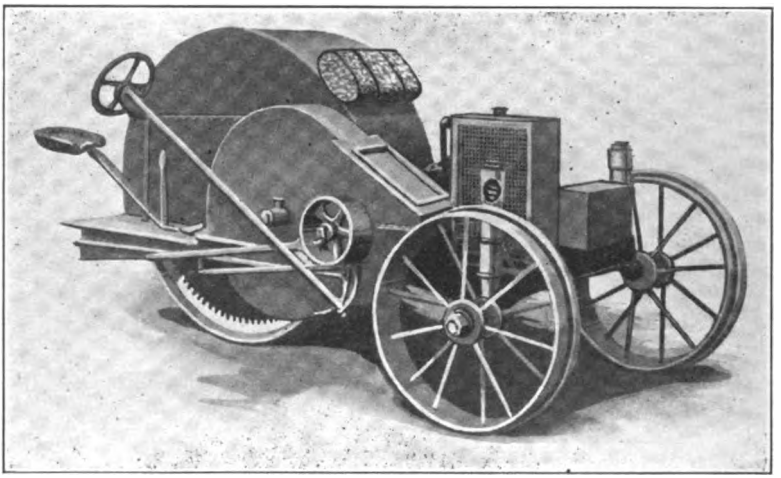

FIG. 23—SIMPLEX TRACTOR DESIGNED TO PULL FOUR-BOTTOM PLOW

the field with tractors and an intimate knowledge of farm conditions.

The foregoing statement must not be taken as a reflection on the enormous amount of work already done by the tractor designers who have been in the field for the past fifteen or twenty years. The pioneer tractor builders who foresaw the need for additional power on the farm deserve great credit for the work already accomplished. While some of the tractors produced do not measure up to the highest standard of engineering practice, there are many good and logical reasons for the types developed and the constructions used.

It was first necessary to demonstrate the possibilities of using tractors instead of horses and this in spite of its apparent simplicity

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has been a tremendous obstacle toward rapid development. It has been necessary also to meet the limitations of a needlessly low selling price in order to overcome a high selling resistance. Like any other new development the tendency in tractor design is toward better materials, greater refinement and higher efficiency.

The farmer who is buying a tractor is more concerned about getting the ability to do his farmwork than about getting a machine. He does not care much what a tractor looks like or what it consists of, or at present even what its efficiency is. He wants something that will do his work better than a horse, enabling him to get his crops in and make more money.

REVIEW OF TRACTOR APPLICATIONS

Gas tractors, as originally developed, were designed to replace the steam tractor, Fig. 1, used for threshing. About twenty-five years

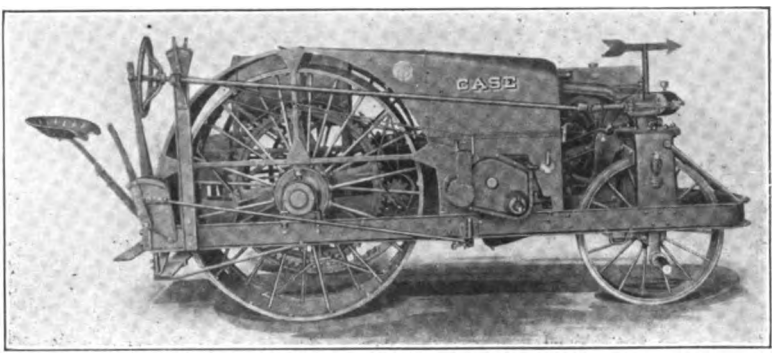

FIG. 24—J. I. CASE T. M. CO.'S 10-20 TRACTOR

ago the steamer, Fig. 2, has been applied to plowing in the prairie wheat country. Most of the heavy gas tractors, such as the Rumely, the International Harvester, Hart-Parr, Case, Kinnard and others, are well known. The C. L. Best machine, Fig. 3, was first made with round wheels and later changed to the caterpillar type, Fig. 4. The Aultman-Taylor, Fig. 5; the "Big Four 30," Fig. 6, made by the Emerson-Brantingham Implement Co.; the Avery, Fig. 7; and the Holt Caterpillar, Fig. 8—these are representative of the earlier developments.

Among the large tractors the four-wheel type predominates. Engine types vary considerably, there being two- and four-cylinder horizontal and two-cylinder opposed; these various horizontal types are used to about the same extent as the four-cylinder vertical type.

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Six-cylinder 125 to 150-hp. engines are used on some of the largest units.

The large tractors as a class are particularly adaptable to threshing and plowing wheat and the other small grains. On large farms several hundred or thousand acres in extent these tractors, with a ten- to fourteen-bottom engine gang-plow, can each turn over 25 to 30 acres per day. They are used successfully for rolling and disking the newly turned sod and to some extent for seeding, which is accomplished in much the same manner as disking. These tractors are also used to pull a number of binders, to run a grain separator and to haul the grain across country to the nearest railroad station.

In sandy ground, where it is difficult to get traction with the round wheel, the endless-track type tractor is frequently used. The builders of this type claim that it is suitable for any work that the round-wheel tractor of similar size can do and that it will operate under

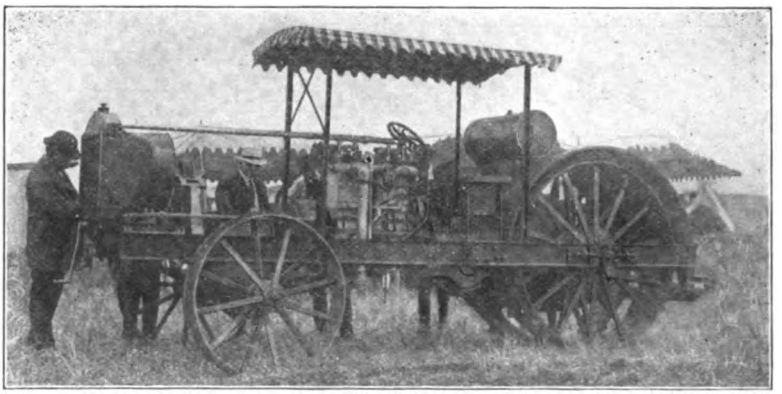

FIG. 25— KINNARD TRACTOR, MADE BY KINNARD-HAINES CO.

many conditions where the round-wheel type cannot. In the wheat country of the Northwest both the round-wheel and the caterpillar-type tractors are used extensively to operate the huge combined harvesters. About twice as much work is done with the tractor of 30 drawbar horsepower as with the animal-drawn harvester, even when thirty-three mules are used with the latter. Large tractors, of both the round-wheel and the endless-track type, are used extensively for road construction, grading, hauling gravel and for work on drainage or irrigation projects. Equipped with special trailers they are used for long-distance contract hauling.

The tractor has also been of great value in the present war in hauling heavy siege guns. A new tractor industry has grown up in

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Germany since the war began. Some five thousand of one particular type of machine (which, before the war had only been developed experimentally) have been sent into Russian Poland and set to work to increase the food supply. These tractors are capable of handling about seven plows; they are equal in power to some of the larger sizes that we have made over here, but are of a lighter and more efficient construction.

SMALL ONE-MAN MACHINES

The machines, Figs. 1 to 8, represent what might be termed the first step in the development of farm tractors. The next step was to reduce the size of some of these large units, make them adaptable to smaller acreages and capable of being handled (with plows) by one operator. Tractors in this class are principally four-wheel types,

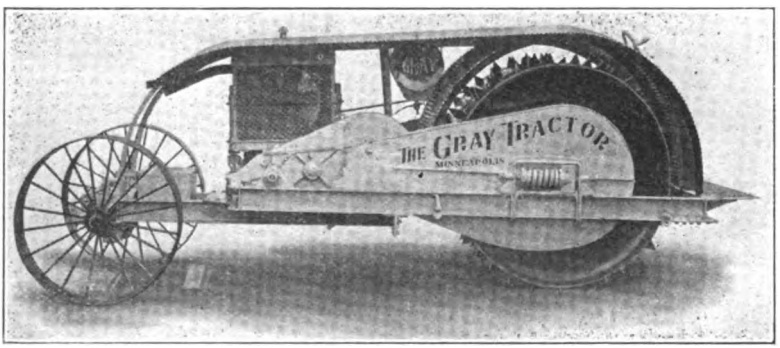

FIG. 26—GRAY TRACTOR, MADE BY GRAY TRACTOR MFG. CO.

ranging in size from six-plow down to two-plow capacity. To show the range of engine sizes and capacity of various tractors I will give the important dimensions for several of them.

The Case "12-25," Fig. 9, was one of the first tractors developed for this class of work and is normally rated at four-plow capacity. It has a two-cylinder opposed 7 by 7-in. engine, a two-speed enclosed transmission, and weighs about 9000 lb.

The Emerson "Big Four," Fig. 10, has a four-cylinder 5 by 7-in. vertical engine and an enclosed transmission, Fig. 11. It pulls five plows normally, and weighs about 10,000 lb. complete with plows, which are attached directly to the tractor frame. The plows can be elevated by a derrick arrangement at the rear, which makes a unit of the tractor and plows.

The Wallis "Cub," Fig. 12, has a four-cylinder 6 by 7-in. vertical engine, a unit powerplant in which is enclosed the engine, two-speed

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transmission and jackshaft. It has a single front wheel, Fig. 13, and is rated at 4000-lb. drawbar pull, which is sufficient capacity for six plows under average conditions and for four plows under practically all conditions. This machine is mounted on springs, front and rear, and weighs about 8500 lb.

The 45-hp. Holt Caterpillar, Fig. 14, has no front wheel, the steering being accomplished by independent control of the endless tracks. It has a four-cylinder 6 by 7-in. vertical engine and a two-speed enclosed transmission. The normal capacity is six plows and it weighs about 12,000 lb.

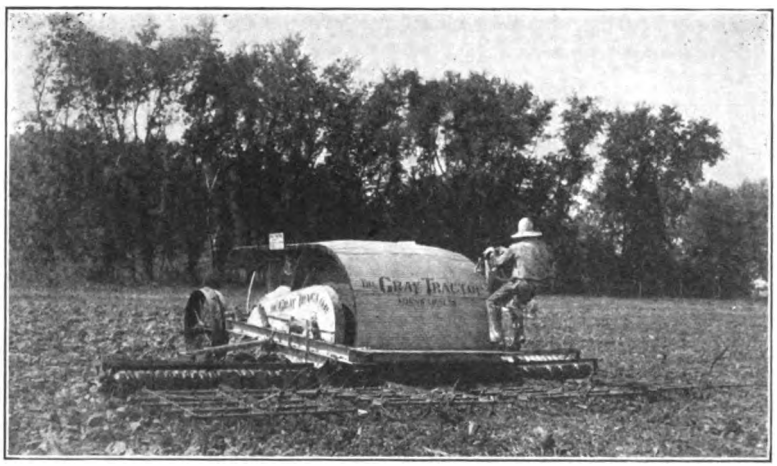

FIG. 27—GRAY TRACTOR PULLING PACKERS AND HARROWS

The Mayer "Little Giant" is built in two sizes, both alike as to general design. The larger, Fig. 15, has a four-cylinder 5¼ by 6-in. vertical engine. The weight is 8700 lb. and it is rated at 4000-lb. drawbar pull. In this machine the engine is considerably smaller than in the two previous; this is accounted for by the fact that it is a higher speed engine.

The smaller model Mayer has a four-cylinder 4¼ by 5-in. vertical engine, and is rated at 2000-lb. drawbar pull. It weighs about 5200 lb. The powerplants, Fig. 16, of both models are identical in design and consist of the engine, a three-speed transmission and the jackshaft, all enclosed in cast housings, which are bolted together to form one unit. This powerplant is carried in a light channel frame, with both front and rear axles mounted on springs.

The Lauson tractor, Fig. 17, which has just been placed on the

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market, has a four-cylinder 4 by 6-in. vertical engine and an enclosed two-speed transmission. It is rated at three-plow capacity. The weight is about 5500 lb.

The William Galloway "Farmobile," Fig. 18, has a 3¼ by 5½-in. engine and a unit-type transmission, which is in front rather than in back of the engine. The final drive is by chain. This machine is rated at three plows and weighs 5000 lb.

FIG. 28—TWO-WHEEL LION TRACTOR PULLING TWO PLOWS

The "Waterloo Boy" tractor, Fig. 19, has a two-cylinder 6 by 7-in. horizontal engine and is rated at two to three-plow capacity. It has a simple one-speed transmission, all exposed, and weighs only about 4800 lb. This tractor is particularly useful for stationary work.

The International "8-16" tractor, Fig. 20, has a single-cylinder 10 by 12-in. horizontal engine, which is hopper cooled. The transmission

FIG. 29—ALLIS-CHALMERS CO. LIGHT FARM TRACTOR

is of the planetary type, giving forward and reverse speeds of about 2 m.p.h. The final drive is by single chain. It pulls two or three plows and weighs about 5000 lb.

The machines just shown are fairly representative of the type of

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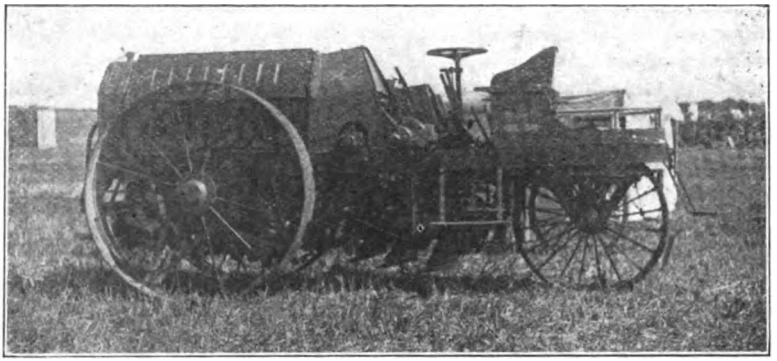

FIG. 30—HACKNEY TRACTOR OR MOTOR PLOW

tractor designed primarily as a pulling machine; they also have considerable usefulness for belt work up to the capacity of the engine. Numerous types have been developed to meet some special requirements of operation, to reduce the cost of production and to overcome some of the limitations caused by reduction in the size of the power-plant.

TRACTORS WITH SINGLE DRIVE-WHEEL

One of the first tractors of a special type of construction to be placed on the market in considerable quantities was the machine, Fig. 21, made by the Bull Tractor Company of Minneapolis. The principal considerations in developing this type were to center the pull in line with the plows, thus eliminating side draft, to arrange the wheels so

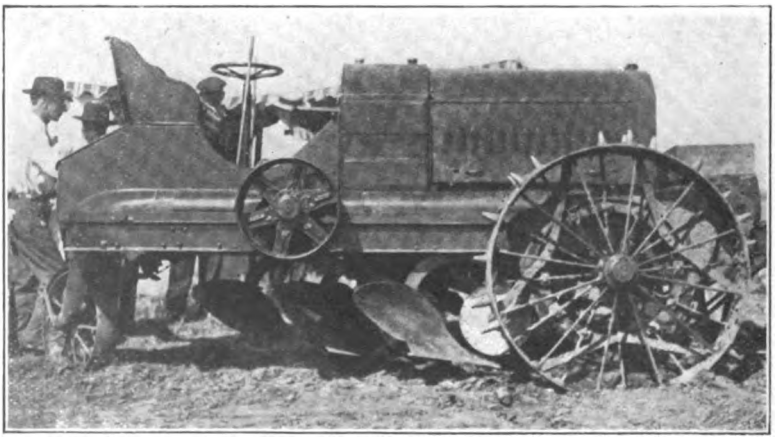

FIG. 31—LAWTER TRACTOR WITH PLOWS UNDERNEATH

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that the machine would in a measure be self-steering, and to simplify the whole design so that the machine could be sold at a low price. This tractor has a two-cylinder opposed engine, spur-gear primary reduction and a roller or lantern-pinion final drive to the single driving-wheel.

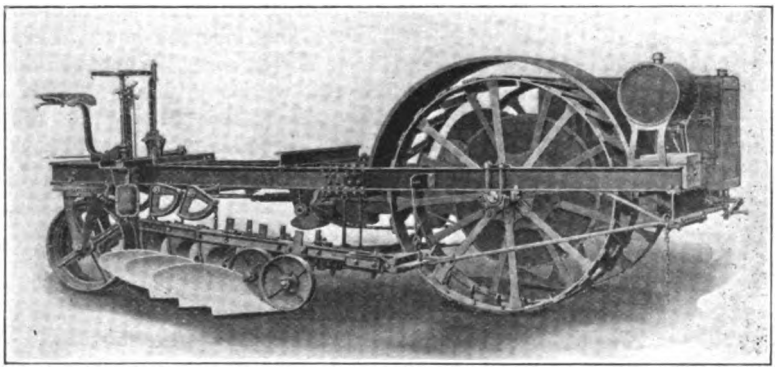

FIG. 32—ADVANCE-RUMELY CO., 12-24 ALL-PURPOSE TRACTOR

Another type, Fig. 22, the Emerson model "L," has a single driving-wheel. The front axle is so arranged that the front wheels are in line; one of the front wheels is made adjustable vertically

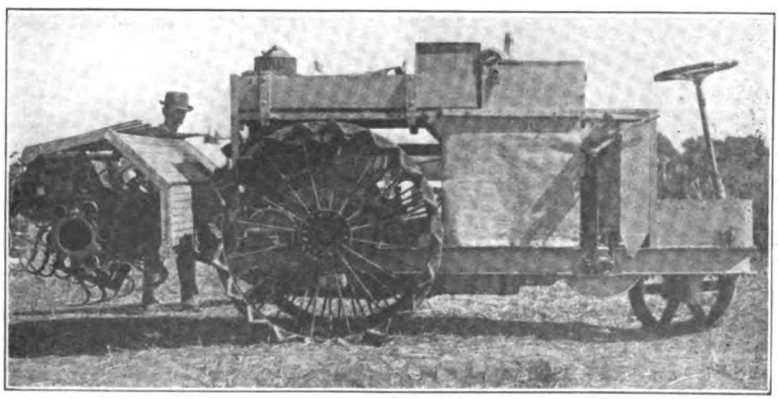

FIG. 33—SOIL MILLER MADE BY ALLIS-CHALMERS CO.

so that it can be run in the furrow, making it practically self-steering. This machine has a four-cylinder vertical engine, set longitudinally, with an inclosed transmission having two speeds forward and reverse.

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A modification of this type, Fig. 23, has been developed by the Simplex Tractor Company. This machine has a double opposed engine, set crosswise, and an enclosed two-speed transmission. It is designed for pulling four plows and is probably the largest tractor of the single drive-wheel type.

In order to overcome some of the limitations of a single drive-wheel tractor, and still eliminate the differential, the Case company brought out its "10-20" model, Fig. 24. This machine has two rear wheels of equal diameter, mounted on a live axle. When running straight ahead the narrower of the two wheels is engaged with a jaw clutch and driven positively by the axle that carries the wide drive-wheel. The steering wheel ahead is practically in line with the

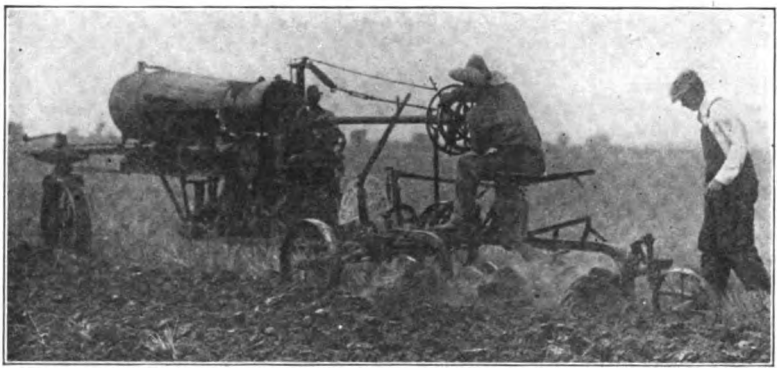

FIG. 34—BATES STEEL MULE WITH ENDLESS-TRACK DRIVE

center of draft, which is located a correct distance from the edge of the wider driving-wheel to enable the machine to run on the unplowed land at all times and still operate successfully, without side-draft, a two or a three-bottom plow. This tractor has a vertical four-cylinder engine set just ahead of the main drive-wheel and crosswise of the frame.

Another tractor that eliminates the use of a differential is the Kinnard, which has two drive-wheels mounted close together, with a single-chain final drive. This machine, Fig. 25, has a capacity of four plows. The total width over the drive-wheels is such that the center line of draft from the plows coincides with the center line of the tractor. One front wheel is adjustable vertically so that it can run in the furrow.

This type of design (without differential) was carried still further in the machine, Fig. 26, developed by the Gray Tractor Manufacturing Company. This has a single drive-drum slightly wider than the plows being pulled. The principal consideration was to distribute the weight

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over a wide area so that the tractor would roll the land down uniformly. This type of construction has a considerable advantage in plowing corn-stubble fields, since the standing corn-stalks and other vegetation are rolled down ahead of the plow and can be thoroughly covered. In harrowing, packing and other similar operations the drive-drum takes the place of one section of packer.

TYPES WITH TWO DRIVE-WHEELS

In order to dispose all the weight on the drive-wheels and obtain the greatest possible traction with the least weight, the two-wheel

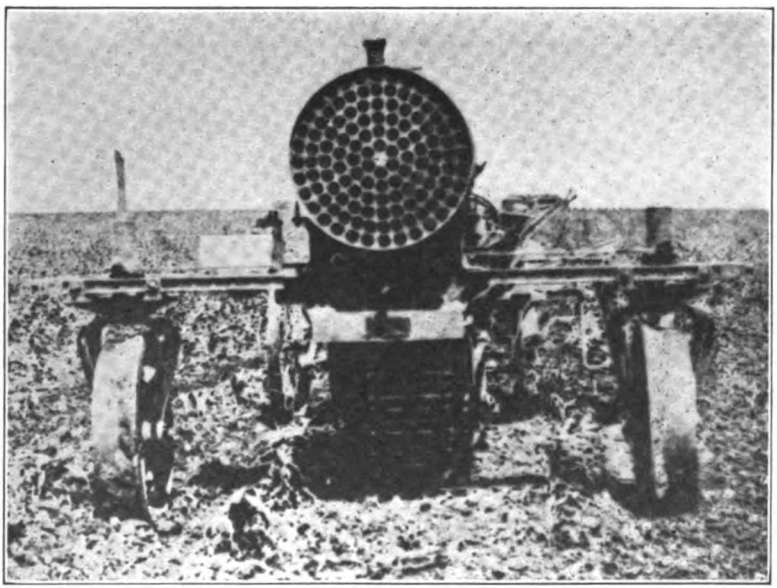

FIG. 35—FRONT VIEW OF BATES STEEL MULE

Lion tractor, Fig. 28, was developed. In operation one of the wheels runs in one furrow and the other on the unplowed land. To facilitate steering differential brakes are applied, one on either jackshaft, these being controlled independently for short turning. In ordinary plowing steering is accomplished with the small caster wheel at the rear.

The Allis-Chalmers tractor, Fig. 29, is a modification of this type. The front (steering) wheel is mounted ahead and on a line with one of the rear wheels so that the one front and one rear wheel run in the furrow. Both rear wheels however are driven. The Lion and the Allis-Chalmers machines have two-cylinder opposed engines and simple gear-reduction transmissions.

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In order to facilitate control of the plows, machines with the plows mounted directly underneath the body of the tractor have been developed by the Hackney Company, Fig. 30, and the Lawter Tractor Company, Fig. 31. The drive-wheels are ahead of the plows and the steering is accomplished by the caster wheel at the rear. In the Lawter each wheel is driven by an independent planetary reverse gear, which makes it possible to drive one wheel ahead and one wheel backward, thereby pivoting on the center of the machine.

A modification of this design, Fig. 32, was recently brought out by the Rumely Company. In its machine however there is but one wide-face drive-wheel running ahead of the plows. The engine is carried between the drive-wheel and an idler wheel of equal diameter.

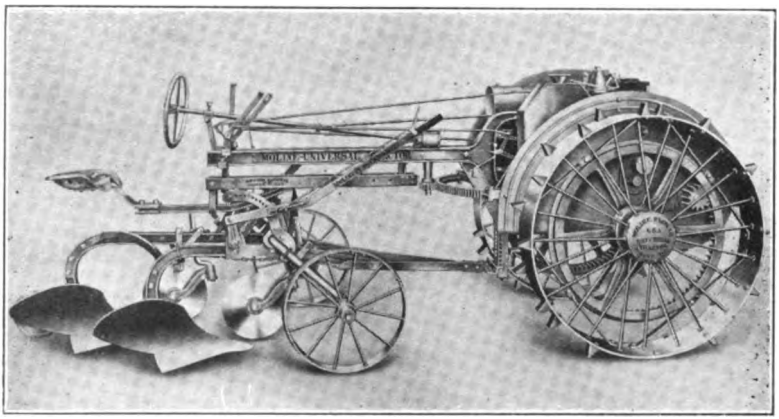

FIG. 36—MOLINE UNIVERSAL TRACTOR AND PLOW ATTACHED

Steering is accomplished by the small wheel at the rear. The transmission and steering control are so worked out that the machine will run with equal facility in either direction. When it is desired to use this tractor for work other than plowing, the plows are detached, the operator's seat reversed and the vertical steering-wheel, drills, harrows or other implements pulled from the drawbar, which is normally forward of the drive-wheel.

In order to make the application of mechanical power more direct, a soil miller, or rotary tiller, has been developed. The one shown in Fig. 33 is made by the Allis-Chalmers Company. A revolving drum carrying teeth or claws is provided at the rear of the machine; its action on the soil is similar to that of a milling cutter or circular saw. This machine thoroughly pulverizes a seed bed to a given depth. It performs at one operation all of the work done by the ordinary mold-board or disk plow, clod crushers and harrows. With a suitable supplementary attachment grain can be planted in the same operation.

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With the exception of the soil miller all of the tractors previously shown are primarily pulling machines to be used in preparation of the soil and in planting or harvesting operations. These machines are intended only to supplement the use of horses on the farms, since they cannot cultivate row crops, except to a limited extent. It is conceivable that these tractors could be used without the assistance of horses on a strictly small-grain farm, but not on farms having mixed production of corn and other row crops in addition to small grains.

To meet the requirements of the corn-belt farmer a different type of machine is necessary. The Bates "Steel Mule," Fig. 34, has a single

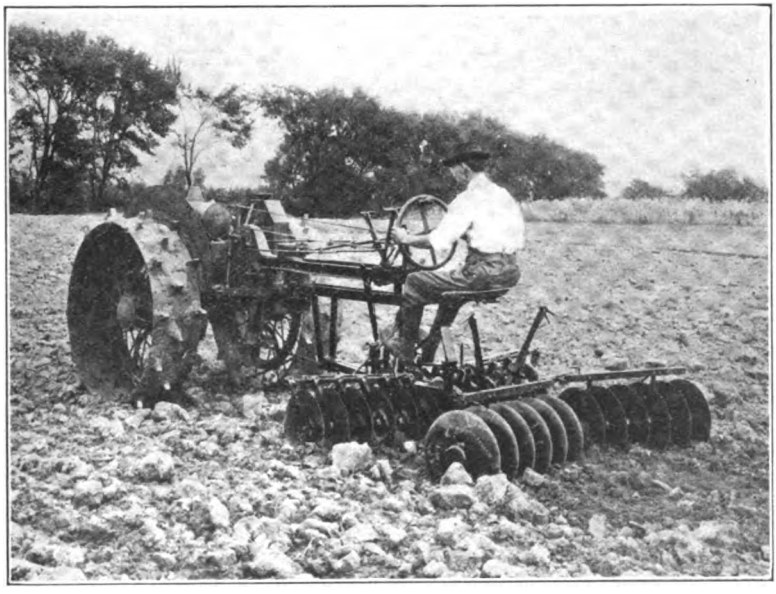

FIG. 37—MOLINE UNIVERSAL TRACTOR OPERATING DISK HARROW

MACHINES FOR CORN-BELT FARMERS

endless-track drive at the rear, narrow enough to run between the corn rows. The front wheels are wide enough apart and are adjustably mounted crosswise so that they can run in the center between the next outside rows (see Fig. 35). The operator can operate the cultivator itself properly because the steering and control devices are carried to the rear of the tractor in the form of a telescoping staff. Any of the horse-drawn implements could be driven in the customary way with the operator seated upon the implement itself. The tractor is hitched ahead of the implement the same as a horse.

Another machine designed to meet a wide range of operating conditions is the Universal tractor, Fig. 36, made by the Moline Plow

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FIG. 39—FARM POWER UNIT MADE BY FAIRMAN MOTOR IMP. CO.

Company. This can be attached to practically all farm implements in the same way as it is attached to a plow, in each case the implement being operated forming the rear support, Fig. 37, for the tractor. The powerplant proper is attached to the implements by a horizontal member provided with a vertical hinge. In steering the whole powerplant is turned in the direction in which it is desired to go. When used for plowing, one of the drive-wheels runs in the previous furrow; to accomplish this and still maintain the level of the machine, this wheel is adjustable vertically. For corn cultivation the drive-wheels straddle one corn row, a high clearance in the center of the machine being provided so that it can be operated as long as it is necessary to cultivate the corn.

The tractor, Fig. 38, made by the Bean Spray Pump Company, is

FIG. 38—PLOWING WITH BEAN SPRAY PUMP CO.'S TRACTOR

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designed primarily for plowing and cultivating in orchards. The entire mechanism is mounted on a single endless track; the overall height is about 3½ ft., the extreme width 34 in. and the length over the steering truck about 6 ft. Steering is accomplished by turning the powerplant, which is attached to the horizontal member by a vertical pivot. The wheel truck shown is detachable, and with properly designed implements the horizontal member can be attached directly to the plows or other tools. On account of its extremely narrow width this tractor will run between the corn rows.

The Fairman Motor Implement Company has developed a machine, Fig. 39, that will run between the rows, but makes use of round wheels for the drive instead of an endless track. The control of this machine is similar to that of the Universal or Bean tractors. It differs from them in that the rear wheels and the supporting frame are an integral part of the machine, and the implements to be operated are either carried on this framework or between the power-

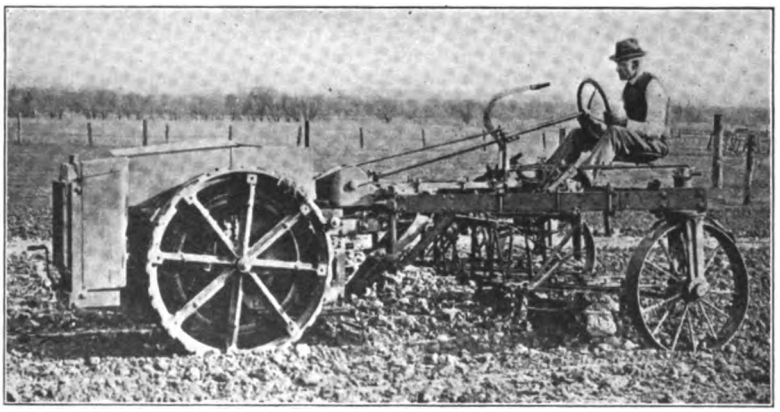

FIG. 40—FAIRMAN POWER UNIT WITH DOUBLE-ROW CULTIVATOR ATTACHED

plant and the rear wheels. For corn cultivation the wheels ordinarily provided for a horse-drawn cultivator are dispensed with entirely and the cultivator framework attached directly, Fig. 40, to the horizontal frame-members of the tractor. The rear wheels of this machine can be steered by the operator; this gives him an excellent control of the cultivator shovels. Sufficient power is provided in the engine to make this tractor equal in pulling power to four horses. Practically all of the other farm implements can be attached to the tractor framework, which is arranged to be adjustable in many directions. The wheels and all other parts of existing farm implements, which are primarily required for use with the horse, are dispensed with and only the operating parts used.

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PAPERS PRESENTED AT MEETINGS OF SECTIONS OF THE SOCIETY

CLEVELAND SECTION

CARBURETER INVESTIGATIONS

By Frank H. Ball and Frederick O. Ball

(Members of the Society)

Abstract

The results are given of laboratory investigations made of a number of different types of carbureters, showing the relation between their gasoline and air consumptions over a wide range. This relation is plotted on so-called quality diagrams, on which is indicated the range between which high power and high efficiency can be expected.

A description is given of a carbureter arranged in two stages, the first being used at light load and the second coming into action when the throttle is nearly open, thereby more than doubling the carbureter capacity. Engine performance curves are presented showing the result when only one or both stages of this carbureter are used.

The laboratory used in the investigations reported in this paper contained a gasoline engine and electrical equipment for loading the engine; also a metering outfit for measuring the gasoline and air used by the carbureter under all conditions. A steam injector was used as another means for drawing air and gasoline through the carbureter.

A water column provided the means for observing the suction or head that impelled the flow, thus furnishing the data necessary for plotting curves showing the quantities of gasoline and air discharged at each end.

FLOW OF AIR AND LIQUIDS THROUGH ORIFICE

Contrary to prevalent opinion, the same law governs the flow of air and of gasoline through a fixed orifice. The quantities discharged in both cases vary as the square root of the suction or head. This law does not hold good at extremely high velocities, but can be accepted as approximately correct throughout the range covered in carbureter practice.

It follows therefore that in a fixed-orifice carbureter the ratio of the flow of gasoline and air at any point will be the ratio throughout the whole range of working capacity.

In practice it is not desirable to maintain the gasoline level at the overflow point in the discharge nozzle; therefore gasoline and air flow do not coincide. This is illustrated in Fig. 2. These curves indicate that with a suction of 20 in. of water the capacity of the respective orifices was such that 10 oz. of gasoline and 100 cu. ft. of air were discharged per minute. This ratio has been found to give a strong

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clean-burning mixture. It represents approximately 12 lb. of air to 1 lb. of gasoline. The air and gasoline curves in Fig. 2 do not have a common zero and therefore the ratio of gasoline to air is not constant.

The actual ratio of gasoline to air, under the conditions shown in Fig. 2, is represented in Fig. 1. The right-hand end of the quality

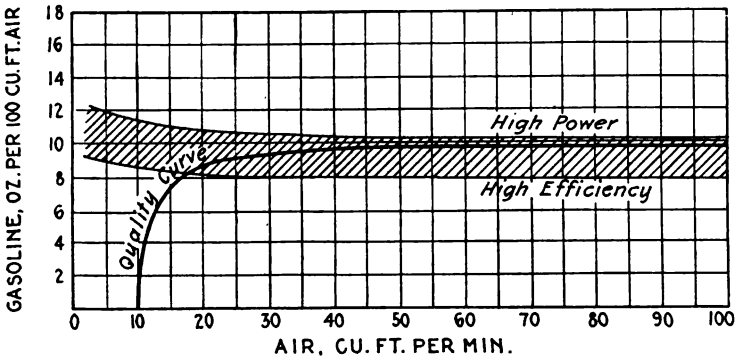

FIG. 1—QUALITY DIAGRAM FOR ELEMENTAL FIXED-ORIFICE CARBURETER

curve indicates 10 oz. of gasoline per 100 cu. ft. of air and corresponds with the right-hand end (where they coincide) of the quantity curves in Fig. 2. Toward zero, the quality curve of Fig. 1 shows the effect of the difference in the zeros of the two quality curves of Fig. 2.

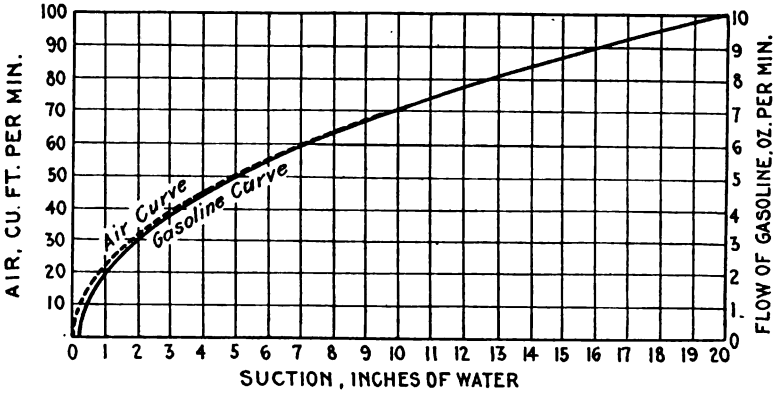

FIG. 2—AIR AND GASOLINE FLOW FOR ELEMENTAL FIXED-ORIFICE CARBURETER

The shaded zone in Fig. 1 covers the range within which it has been found desirable that the quality should be maintained by the carbureter. The boundaries of this zone may not be accepted universally as the best, but they are approximately the best. The zone is

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entirely satisfactory as a field in which to plot the characteristic quality diagrams of various types of carbureter.

ELEMENTAL FIXED-ORIFICE CARBURETERS

When less than 18 cu. ft. of air per minute is drawn through the carbureter, the quality curve, Fig. 1, drops below the desirable zone. This makes it impossible to enrich small quantities of air. Therefore,

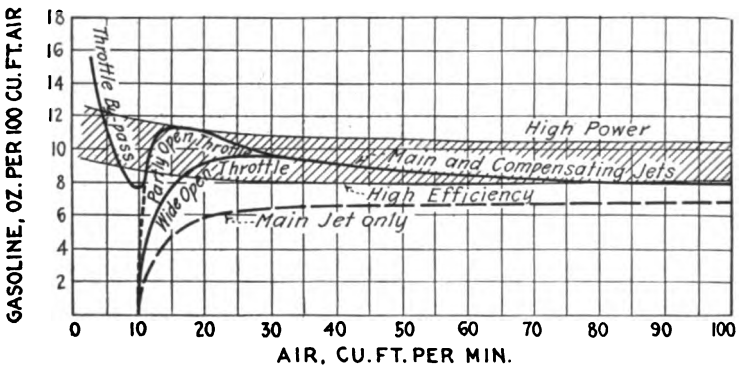

FIG. 3—QUALITY DIAGRAM FOR COMPENSATED FIXED-ORIFICE CARBURETER

the elemental fixed-orifice carbureter cannot be used with success, and it becomes necessary to embody in the carbureter some means for augmenting the gasoline supply where it is deficient. Various devices are used for this purpose. Some effect the necessary compensation by

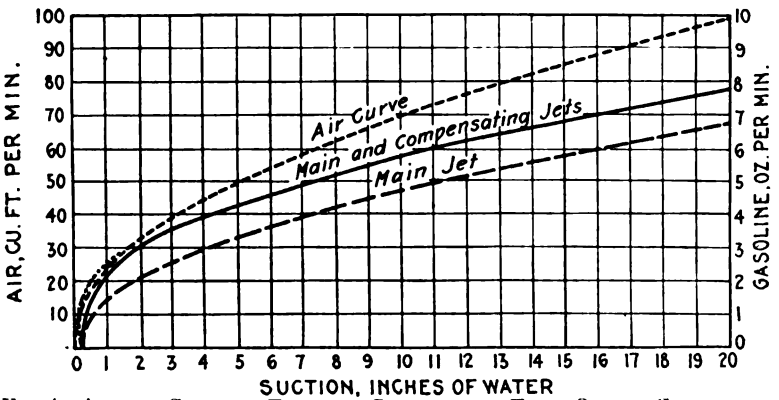

FIG. 4—AIR AND GASOLINE FLOW FOR COMPENSATED FIXED-ORIFICE CARBURETER

acting on the gasoline supply, others act on the air and still others on both.

It is proposed to analyze the effect of the most important of these devices, selecting those which have been used most extensively in

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high-grade instruments. The analysis is made solely to show the effect of the various devices in maintaining a normal quality at a lower minimum quantity than is possible with the elemental fixed-orifice carbureter. No attention is paid to the relative maximum

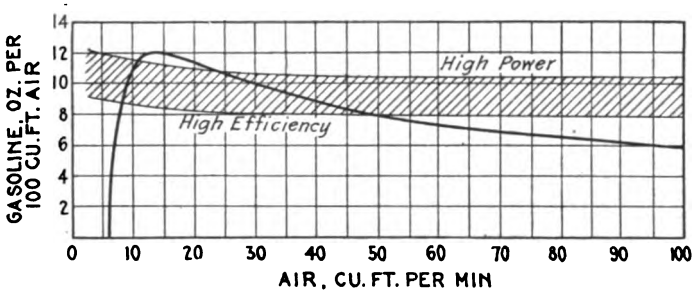

FIG. 5—DIAGRAM FOR ELEMENTAL AIR-VALVE CARBURETER WITHOUT FIXED ORIFICE

capacities or to any other questions relating to the several types. These will be considered later in connection with quality diagrams for the actual carbureters.

COMPENSATED FIXED-ORIFICE CARBURETER

This differs from the elemental fixed-orifice carbureter in that it is provided with compensating devices that increase the supply of

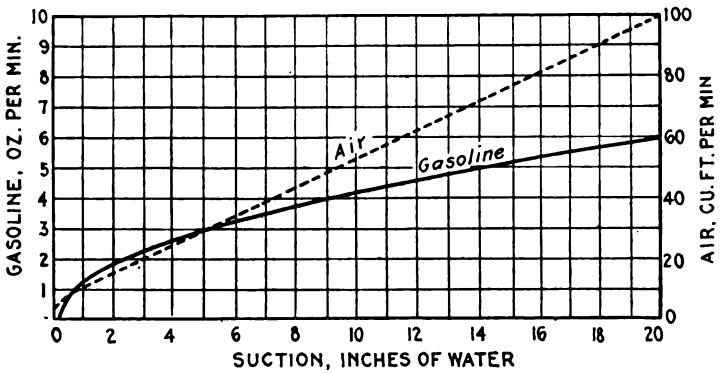

FIG. 6—FLOWS FOR ELEMENTAL FIXED-ORIFICE CARBURETER WITHOUT FIXED ORIFICE

gasoline, particularly where it is most deficient. Figs. 3 and 4 illustrate the effect of these devices. The quantities discharged by the main jet under various amounts of suction are shown by the main-jet quantity curve in Fig. 4. The quality produced by these quantities is shown in Fig. 3 by the curve marked "main jet only."

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A second supply is drawn from a well open to the atmosphere and supplied with a constant quantity of gasoline from the float-chamber by gravity. The total quantities of gasoline from this double supply are indicated by the curve marked "main and compensating jets" in Fig. 4; the effect on the quality is indicated in Fig. 3 by the curve marked "main and compensating jets." The two branches of the latter marked "partly-open throttle" and "wide-open throttle" come together at a common zero. This will be explained later. A third supply of

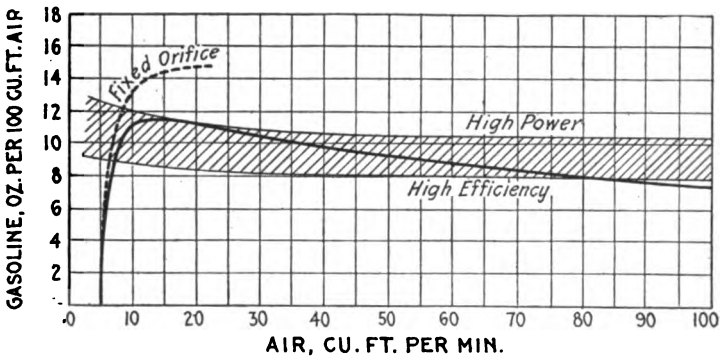

FIG. 7—DIAGRAM FOR SIMPLE CARBURETER WITH COMPENSATING AIR VALVE

gasoline is drawn from the gravity well already referred to and is discharged at a point on the engine side of the throttle. This is also essentially a constant quantity and its effect on the quality is indicated by the curve marked "throttle by-pass" in Fig. 3.

When the throttle is nearly closed the by-pass comes into action and draws gasoline from the gravity well, but when the throttle is wide-open and the engine is slowed down by additional load the

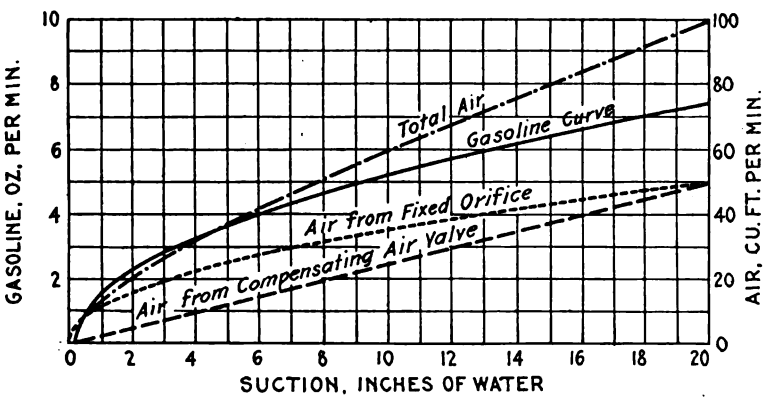

FIG. 8—FLOWS FOR SIMPLE CARBURETER WITH COMPENSATING AIR VALVE

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decreasing suction of the engine permits the well to gradually fill with gasoline, thus decreasing the gravity head in the float-chamber and causing the quantity discharged into the well to decrease. The effect

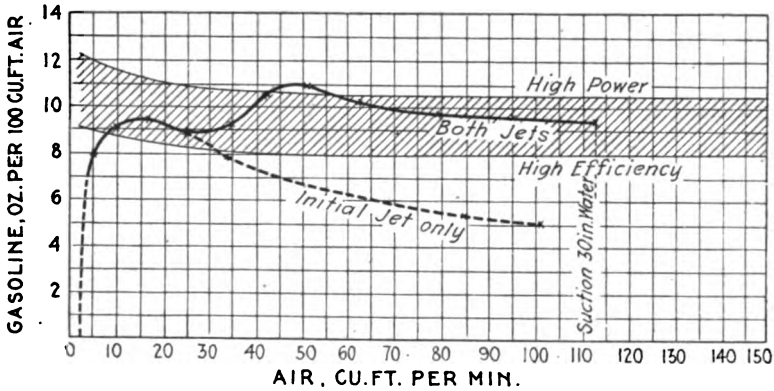

FIG. 9—DIAGRAM FOR ACTUAL AIR-VALVE CARBURETER WITH SUPPLEMENTAL GASOLINE JET

of this smaller quantity of gasoline drawn from the atmospheric well is shown on that part of the quality curve marked "wide-open throttle." The other branch, marked "partly-open throttle," indicates the

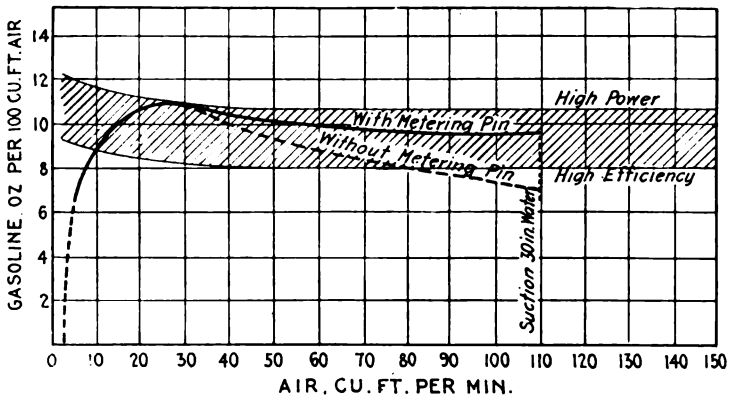

FIG. 10—DIAGRAM FOR ACTUAL AIR-VALVE CARBURETER WITH METERING PIN ATTACHED TO AIR VALVE

quality when, by progressively closing the throttle, the by-pass comes into action and prevents the level in the well from rising.

With this arrangement then the carbureter has two distinct quality curves as shown by the full lines. This completes the analysis of this

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class of carbureter. It is a small class and no further types will be considered.

AIR-VALVE CARBURETERS

This class is much larger than all others combined and is represented by a large number of types. Only a few of the most important will be considered.

The air-valve carbureter will be best understood by first making an analysis of the simplest form, in which all the gasoline is admitted through a fixed orifice and all the air through a valve closed by a spring. This is an imaginary type, which could not be used in practice.

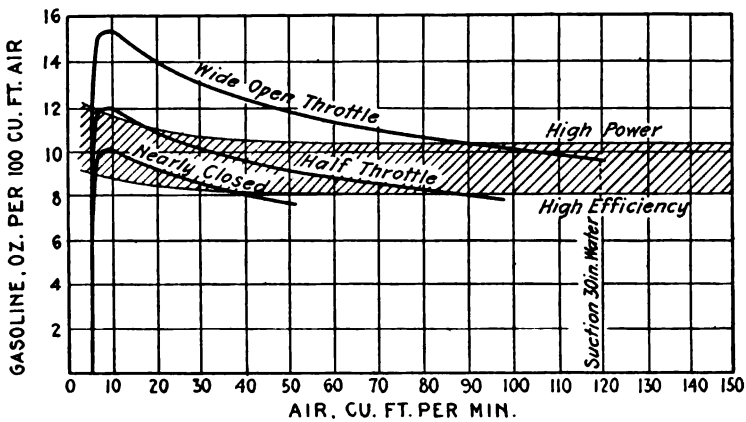

FIG. 11—DIAGRAM FOR ACTUAL AIR-VALVE CARBURETER WITH NEEDLE VALVE CONTROLLED BY THROTTLE

In Fig. 5 the quality curve does not follow the desirable zone, but crosses it at an angle. The range of capacity through which the mixture would fire is thus very limited. The analysis of this quality diagram will be found in Fig. 6.

Comparing the quality diagram, Fig. 5, with that of Fig. 1, it is evident that a combination of the two systems of control would make a better quality diagram than either. This suggests the type of air-valve carbureter that will be next considered.

Simple Carbureter with Compensating Air-Valve and Fixed Orifice

This type is well known in a variety of forms. The fixed air-orifice is often called the "strangle tube" or the "Venturi choke," and ordinarily the gasoline nozzle terminates in this passage. It will be assumed in this, as in the preceding cases, that the gasoline is regu-

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lated by a fixed orifice, or in other words, that the area of the gasoline orifice remains unchanged. A second air passage is closed by a spring-opposed air-valve, which opens as the suction increases.

The three dotted curves in Fig. 8 represent quantities of air passing through the air passages with different degrees of vacuum in the manifold. The lower dotted curve indicates the quantities admitted by the progressively-opening air-valve. The next represents the quantities admitted through the fixed air-orifice and the upper dotted curve is the sum of these quantities and is therefore the total air. The full-

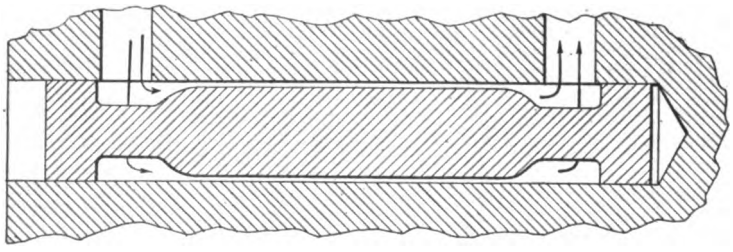

FIG. 12—DEVICE DESIGNED FOR MAGNIFYING LIQUID FRICTION

line curve indicates the quantities of gasoline admitted through the fixed gasoline orifice.

In Fig. 7 the dotted line shows what the quality would have been if the air-valve had been prevented from opening, and the full line indicates the quality when the air-valve is free to open in response to increasing suction.

Air-Valve Carbureter With Supplemental Gasoline Jet

Various means have been used to cause a second gasoline jet to come into action before reaching maximum capacity, thus augmenting the supply when it has become deficient. If the second jet comes into service abruptly, it causes the quality diagram to make abrupt changes of quality. This is shown in Fig. 9, which actually represents the action of a well known carbureter of this type. The dotted line shows what the quality would have been with the first jet only, and the added richness caused by the second jet is shown by the full line.

Carbureter With Metering Pin Attached to Air-Valve

Fig. 10 shows how the effect sought with the double jet (shown in Fig. 9) is produced by a taper metering pin attached to the air-valve and arranged so that when the air orifice is increased by opening the air-valve, the gasoline orifice is simultaneously increased by withdrawing the tapering pin.

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With this device a fluid dashpot attached to the valve is necessary to prevent objectionable reciprocating motion of the metering pin. These moving parts attached to the air-valve prevent a delicacy of action that is desirable when small quantities are being used. The forces acting on the valve are consequently weak. To overcome this difficulty and give stability of action it has been found necessary to make the fixed air-orifice larger than that in the simple carbureter, thus making it possible to keep the air-valve in contact with the seat until the actuating forces shall have become of some magnitude. The effect of this is to make it impossible (see Fig. 10) to enrich a small quantity of air. Under these conditions the closing of the throttle for slow speed reduces the engine speed, partly by reducing the quantity

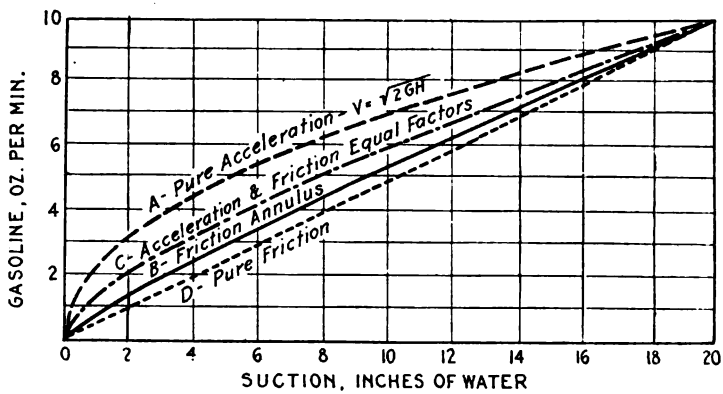

FIG. 13—RELATION OF LIQUID FRICTION TO FLOW OF GASOLINE

of mixture entering the cylinder and partly by decreasing the richness of the mixture. This results in an extremely lean mixture for idling and slow running, which fires with uncertainty and is susceptible to the cold.

Carbureter With Needle-Valve Controlled by Cam on Throttle

Fig. 11 shows quality diagrams for the best known carbureter of this type. It also illustrates the effect produced by this method of control. These curves show that in each case the needle-valve remained in one position throughout, which means that the throttle-valve also remained in one position and the speed was changed by changing the load.

If the load at each speed is such that the necessary position of the throttle causes the needle-valve to supply the amount of fuel required by the air that is being used, then a normal quality of mixture is supplied, but under all other conditions the mixture is either too lean or too rich.

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To understand this clearly, refer to Fig. 11, which shows that when 100 cu. ft. of air is being used, the quality of the mixture is 10. Assuming now that the throttle remains wide-open, and the speed is reduced by increasing the load (as it would be in climbing a hill) the quality grows steadily richer and the engine presently ceases to fire because of excessive richness. This condition is always found when the gasoline orifice is varied by opening and closing the throttle.

FRICTION IN CONTROLLING FLOW OF LIQUIDS

So far the quantity of gasoline flowing through a fixed orifice has been assumed to vary as the square root of the suction or head. The curve representing these quantities is also the curve representing the velocity of a mass in equal increments of time when acted on by a constant force, such as the force of gravitation, and the law is called the "law of acceleration." The friction of liquid flowing through an orifice is therefore a negligible quantity. The inertia of the liquid opposing acceleration through the orifice is practically the only factor to be considered and the flow can be called an "acceleration flow." Liquid friction can, however, be so magnified relatively to the inertia of the liquid as to become the dominating factor. The shape of the curve is

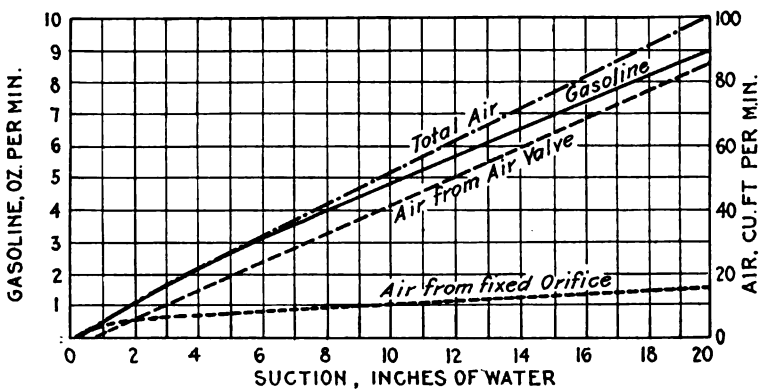

FIG. 14—FLOWS FOR AIR-VALVE CARBURETER WITH FRICTION CONTROL OF GASOLINE

then greatly changed because the resistance of friction varies directly as the velocity and not as the square of the velocity. Fig. 12 represents a device for magnifying liquid friction relatively to the resistance of its inertia. The liquid passes through a thin annulus of relatively large diameter. The thin liquid tube encounters friction on a large surface, both inside and outside. Because of the high friction thus developed, the cross-sectional area of the annulus must be much larger for a given quantity than would be required with an orifice flow, and this decreases the velocity and consequently the effect of inertia. The rounded corners of the entrance to the annulus eliminate a resistance called "entrance head."

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The flow of gasoline is represented by the curves in Fig. 13. The curve *A* represents the discharge through an orifice. The curve *D* connecting the ends of *A* represents the discharge that would take place if friction were the only resistance to the flow. The flow would then increase directly as the head. This condition is, of course, impracticable. Curve *B* represents the actual discharge when the flow is resisted by a device such as is shown in Fig. 12. The curve *C* passes midway between curves *A* and *D*. All curves between *A* and *C* can be called curves of "acceleration flow," because in every case acceleration is the dominating factor; curves between *C* and *D* can be called curves of "friction flow," because in every case friction is the dominating factor.

Air-Valve Carbureter With Friction Gasoline Control

Fig. 14 shows the relation of air and gasoline flow to suction for a carbureter using friction control of gasoline. This analysis follows the same general plan that was followed on the preceding pages, and the resulting quality diagram is given in Fig. 15.

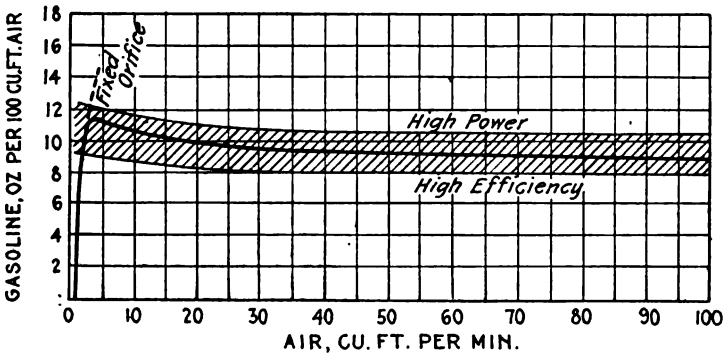

FIG. 15—DIAGRAM FOR AIR-VALVE CARBURETER WITH FRICTION CONTROL OF GASOLINE

It is evident from Fig. 14 that for use with a straight-line air curve, a gasoline flow represented by curve *B*, Fig. 13, is more desirable than that represented by curve *A*. This is true because the quantities of gasoline discharged by a fixed orifice are found (Fig. 6) along a curve, while the quantities of air discharged by a spring-opposed valve are found along a straight line, or nearly so. As a result a constant ratio of air and gasoline cannot be maintained.

COMPARISONS OF QUALITY DIAGRAMS FROM ACTUAL CARBURETERS

The preceding pages have been devoted mainly to the theoretical diagrams relating to several types of carbureter. It is now proposed to investigate the diagrams showing results actually produced by these instruments. Comparisons will be made as to: uniformity of quality;

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maximum capacity (for high speed); minimum capacity (for slow speed); and working range.

A desirable diagram is one that runs parallel to and does not cross the zone of best quality. If the quality diagram indicates rich or lean mixtures, trouble will result under varying atmospheric conditions.

The diagrams were all made from 1¼-in. carbureters. This is the size generally used on engines of from 275 to 300 cu. in. piston displacement. Such engines, if properly designed, require for maximum power at high speed a carbureter capacity of about 150 cu. ft. of air per minute with a manifold vacuum of 30 in. of water.

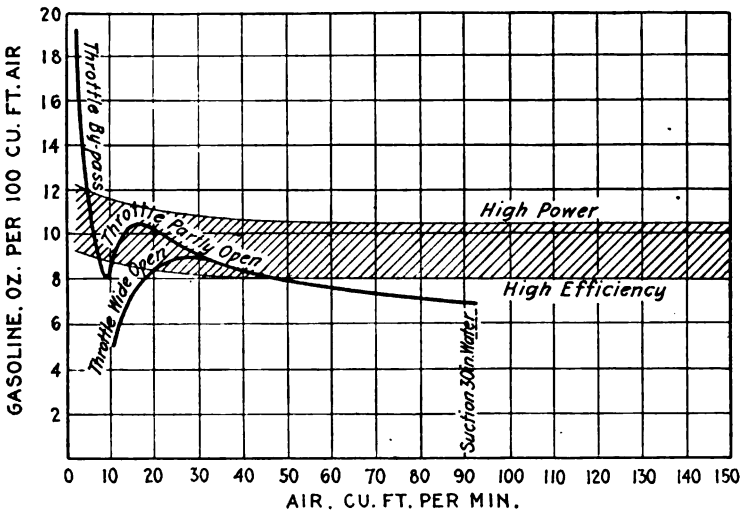

FIG. 16—DIAGRAM FOR ACTUAL COMPENSATED FIXED-ORIFICE CARBURETER

The maximum capacity means little unless considered in connection with the minimum capacity because a carbureter with limited working range can show large capacity, when good performance is sacrificed at slow speed.

To obtain the best slow running and idling, these engines should be supplied with less than 4 cu. ft. of air per minute, and this must be enriched nearly to the maximum. A lean mixture is not desirable when idling as the engine is then susceptible to changes of temperature; an excessively rich mixture is also faulty in that it makes soot on the valves and spark-plugs and loads up the intake passages.

For the purpose of comparison a uniform standard has been adopted for the working range. In each case the maximum capacity is determined by measuring the quantity of air the carbureter delivers with a manifold vacuum of 30 in. of water. The minimum capacity is located at the point where the curve passes out of the zone of best quality.

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Compensated Fixed-Orifice Carbureter

The quality curve, Fig. 16, crosses the zone three times. At high speed the quality is too lean for high power. At 16 to 20 ft. the quality is too rich for high mileage. At 8 ft. there is a lean spot, and at 3 to 4 ft. the mixture is too rich for good idling. The working range is from 4½ to 92 ft., or a ratio of 1 to 20. The capacity is too small for high power.

Simple Carbureter With Compensating Air-Valve

The quality curve, Fig. 17, crosses the zone twice. At 90 ft. the quality is too lean for high power. From 10 to 30 ft. it is too rich for high mileage. The working range is from 6 to 90, or 1 to 15. The capacity is too little for high power (even if the quality be correct).

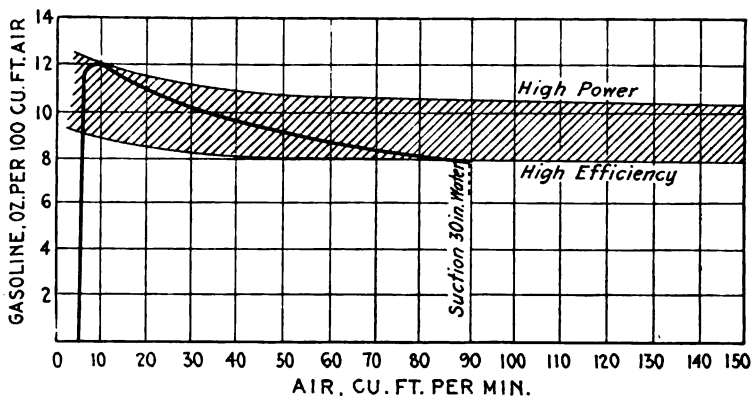

FIG. 17—DIAGRAM FOR ACTUAL SIMPLE CARBURETER WITH COMPENSATING AIR VALVE

Air-Valve Carbureter With Supplemental Gasoline Jet

The quality, Fig. 9, zigzags badly, and while good at 114 ft., is much too rich at 50 ft. and too lean at anything less than 8 ft. The power would be fairly good at high speed, but when using from 40 to 60 ft. of air, the mixture would be much too rich for good performance. The mileage would be low and there would undoubtedly be "loading" in the intake passages. The idling and slow running could not be good as the minimum amount of air enriched is not small enough. The working range is from 8 to 114 ft. or 1 to 14.

Air-Valve Carbureter With Metering Pin Attached to Air Valve

The quality, Fig. 10, does not, in this case, stagger across the zone and for part of its range is correct for high power. At from 20 to 40 ft. (where most of the ordinary running occurs) the quality is too rich for good mileage and "loading" will occur. This can be

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corrected by a different adjustment, but it must be done at a further sacrifice at slow speed, which is already faulty.

The mixture goes out of the zone at 10 ft. and would go out at 15 to 20 ft. if the quality were made efficient at 25 ft. The working range is from 10 to 110 ft., or 1 to 11, and would be much less if an efficient mixture were used at ordinary speeds.

Air-Valve Carbureter With Needle-Valve Controlled By Throttle

The quality, Fig. 11, of the mixture with this carbureter depends on the position of the throttle-valve. If the throttle is skillfully handled, the quality stays well within the zone. If, however, the throttle

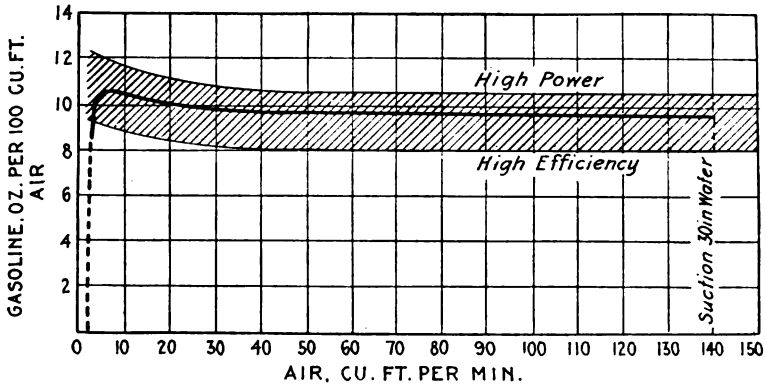

FIG. 18—DIAGRAM FOR ACTUAL AIR-VALVE CARBURETER WITH FRICTION CONTROL OF GASOLINE

is not handled well, the quality will go outside the zone, being both too rich and too lean.

The working range is fairly good—5½ to 120 ft., or 1 to 22. The maximum power will be good and the idling fairly good, but this instrument will load badly when pulling at medium or slow speeds with open throttle.

Air-Valve Carbureter With Friction Control of Gasoline

The quality curve in Fig. 18 follows parallel to the zone of quality throughout its entire range. The working range is from 3 to 140 ft., or 1 to 47. In view of this remarkable showing, it is unfortunate that fluctuations in temperature make so great a variation in the flow of gasoline that the device is impracticable.

Two-Stage Carbureter

(The description of this carbureter is given later.)

The quality curve, Fig. 19, is divided into two parts. Most of the running is done on the part marked "primary stage." The curve follows along the efficient edge of the zone, giving high mileage, and has a low minimum capacity, giving good idling and slow running. When

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both stages are in use the quality is rich enough for maximum power and speed. The two stages are used only for maximum power and will give that power without affecting the mileage obtained at ordinary speeds with the primary stage.

The working range is from 3 to 140 ft., or 1 to 47, and a still greater range can be used successfully, if desired, by increasing the capacity of the second stage. Since the primary stage is independent of the second stage, it is not affected by adjusting the latter.

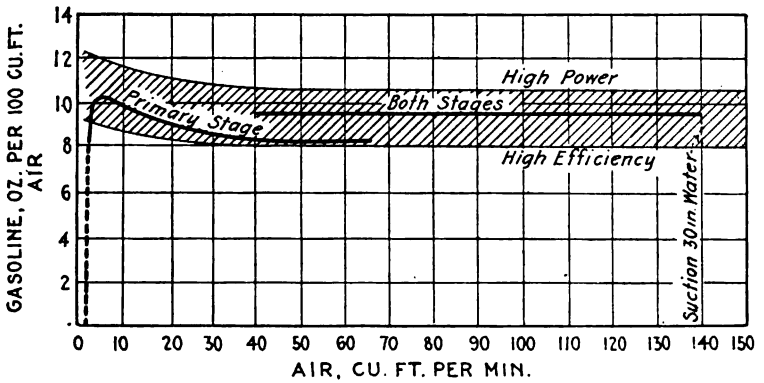

FIG. 19—DIAGRAM FOR ACTUAL TWO-STAGE CARBURETER

CONSTRUCTION OF TWO-STAGE CARBURETER

Many of the difficulties inherent in the types described can be minimized by dividing the carbureter into two stages. Such a device is illustrated in Fig. 20, in which *2* is the primary fixed-orifice for air, and *3* is the gasoline jet located as usual in this passage. A spring-opposed valve *4* controls the flow of air through the valve passage to the mixing chamber. These parts, when connected to a gasoline supply and an outlet to an engine, constitute a simple air-valve and fixed-orifice carbureter, called the "primary stage" of the instrument. Associated with this in the structure is the air passage *5* containing the gasoline jet *6*. These parts, when in action, constitute an elemental fixed-orifice carbureter, such as has already been described. This fixed-orifice carbureter is the "second stage" and generally has more than half the total capacity. Normally the second stage is closed by the butterfly valve *7* and held closed by a spring. A connection to the throttle is so arranged that when the throttle is nearly open, the final full opening throws the valve *7* wide-open, thereby more than doubling the carbureter capacity.

Vacuum Device Gives Quick Acceleration at Slow Speeds

This carbureter has a device, actuated by fluctuations of the vacuum in the manifold, that discharges a predetermined quantity of

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gasoline whenever the throttle is suddenly opened at slow speed. This consists of the plunger *8* having on its upper end an extension *9* which acts as a piston to move the plunger under the influence of the fluctuations of the manifold vacuum communicated through the passage *10*. The plunger is fitted loosely in a cylindrical chamber having a restricted passage at the bottom communicating with the float-chamber so that the level of gasoline in the plunger-chamber is maintained at the float-chamber level. The plunger-chamber has an atmospheric opening *11*, and a passage *12*, to the mixing-chamber.

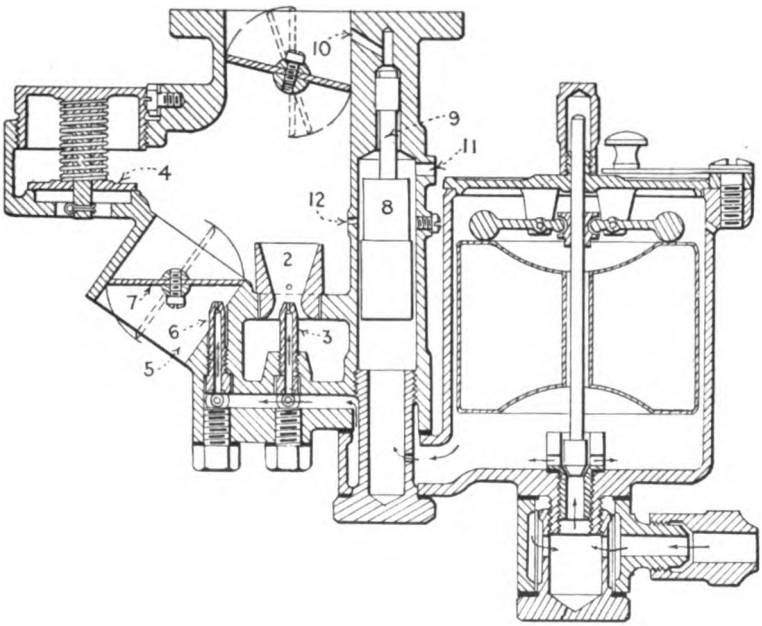

FIG. 20—SECTIONAL VIEW OF TWO-STAGE CARBURETER

The operation is as follows: When the throttle is nearly closed, the vacuum in the manifold lifts the plunger to the position shown, and the space below the plunger fills with gasoline. The device is now ready for action. A sudden opening of the throttle breaks the vacuum in the manifold, which releases the plunger and it drops by gravity, causing the gasoline to pass up to the space above the plunger, where it is swept into the mixing-chamber by the air entering through the passages *11* and *12*. This operation is repeated as often as the throttle is suddenly opened from a nearly closed position.

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Advantages of Two-Stage Instrument

The working range of a two-stage instrument should be greater than that of a single stage because it is the summing up of the ranges of both stages. The atomizing of the primary stage must necessarily be high, even when the total capacity of the instrument is great, because the dividing of a carbureter into two equal stages makes the

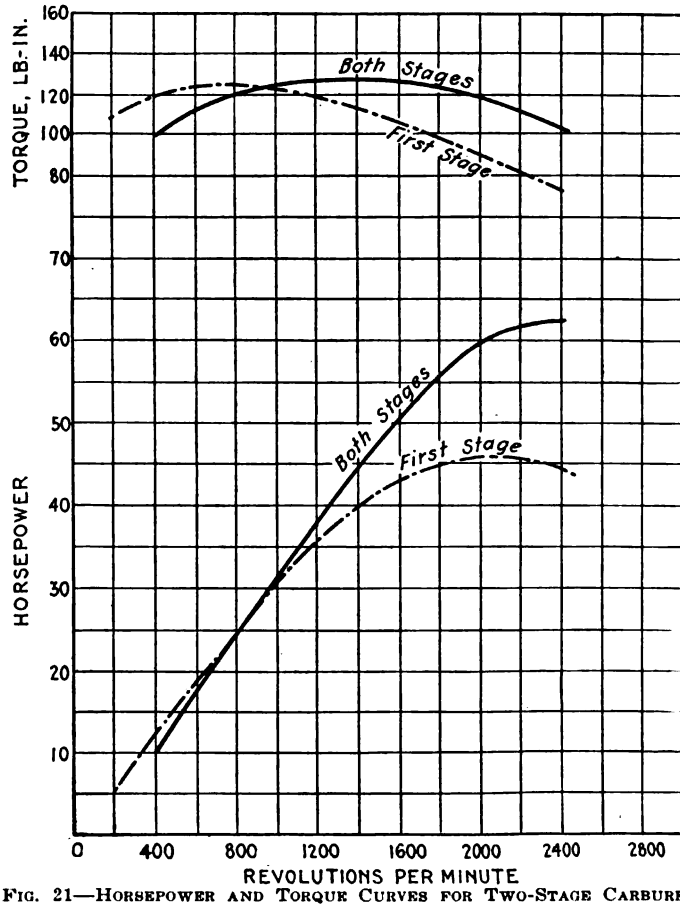

FIG. 21—HORSEPOWER AND TORQUE CURVES FOR TWO-STAGE CARBURETER

atomizing energy of the primary stage four times as great as with the full capacity. This high atomizing facilitates the pick-up to such an extent that it is fairly good practice to make the capacity of the primary stage so small that no pick-up device is necessary. All things considered, however, it is better to divide the two stages more evenly and incorporate a pick-up device.

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The two-stage instrument is peculiarly adapted to obtaining high fuel efficiency, as will be seen by referring to Fig. 19. In this case the setting of the primary stage is such that the mixture is lean. The high atomizing of this stage and the pick-up discharge make this practicable, the result being high fuel efficiency.

Under these conditions, the setting of the second stage is made for a rich mixture so that when it is brought into service for maximum power, the resulting quality from the combined jets will be rich enough for maximum torque.

Inasmuch as more than 90 per cent of the running of the engine is done with the primary stage, the fuel efficiency is practically determined by the quality of mixture used in this stage. The quality diagram of the primary stage differs somewhat from those of the

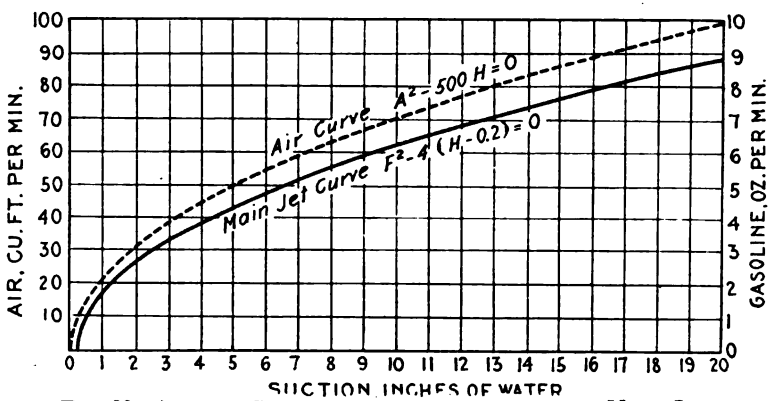

FIG. 22—AIR AND GASOLINE FLOWS WITH DIFFERENT MAIN JETS

simple air-valve carbureters. This is explained by the restricted air passage, or throat, under the air-valve, which becomes a fixed-orifice control of air when the valve has opened slightly. The air-valve simply bridges over from the primary Venturi throat to the largest fixed orifice under the air-valve. This arrangement limits the capacity and could not be used with a single-stage carbureter of normal capacity.

Power and Torque With Two-Stage Carbureter

Fig. 21 illustrates one of the effects of dividing a carbureter into two stages. The torque curve shows this most clearly. At about 900 r.p.m. the torque is the same whether the single stage or both stages are in service. At this speed the loss of atomizing of the large capacity offsets the gain due to low resistance. With the primary stage the gain due to higher atomizing is offset by the higher resistance. Below this speed the primary stage shows the higher torque because of the superior atomizing, and above that speed, the lower resistance of the two stages results in the higher torque. A proper use of the two stages produces in effect a flat torque curve.

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DISCUSSION

V. R. HEFTLER:—In the first part of the paper certain theoretical assumptions are made, which, as stated, are not generally held. The opinion* was presented by Prof. K. Rummel and was contrary to the view of A. Krebs.† Krebs held that the air flow follows a parabolic law, and that the gasoline flow folows the same law, with a slight lag at the start. He attributed the lag to the slight difference in head between the constant level and the tip of the nozzle. Krebs determined the lag experimentally and found the corrective term to be 21 mm. when the nozzle was not 21 mm. above the level.

If the law, upon which all the curves in the paper are based, is true it would be interesting to know how it has been established. The new opinion could easily be tested by eliminating the disturbing influence caused by the nozzle being too high above the gasoline level. We could readily make an experimental carbureter with the nozzle, say ⅛ in. below the level. Such a carbureter would be impractical

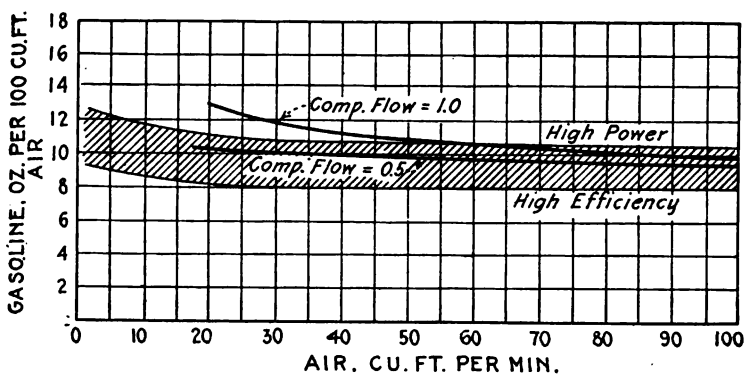

FIG. 23—QUALITY DIAGRAM PRODUCED WITH DIFFERENT COMPENSATING JETS

for an automobile, because it would spill gasoline while at rest, but it could be used to determine why the gasoline curve does not follow the air curve.

JET DESIGN BETTERS PERFORMANCE

But even taking the premises used in the paper in analyzing the various carbureters, it is possible, at least in one case, to arrive at a different conclusion. I refer now to Figs. 3 and 4 for the compensated fixed-orifice carbureter, which is described as follows: "A second supply is drawn from a well open to the atmosphere and supplied with a constant quantity of gasoline from the float-chamber by gravity." This is quoted from a patent granted to Baverey, and therefore refers to only one carbureter (the Zenith).

*See *Horseless Age*, Apr. 7, 1915, p. 474.
†See *Horseless Age*, Apr. 14, 1915, p. 508.

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I have considered the parabolas used in making the curves in Fig. 4 and find that the air curve is accurately represented by the equation $A^2 - 500\ H = 0$, where A is the air flow and H the suction. The fuel curve of the main jet is represented by the equation $F^2 - 2.316\ (H - 0.2) = 0$, where F is the fuel flow. These equations (plotted in Fig. 22) can be verified easily.

I have assumed that the flow through the compensating jet follows a more complex law; that it is constant and gives a quantity of fuel of 1 oz. per minute, provided the suction is greater than 2 in. With lower suctions the flow through the jet is not constant. We shall see later whether these assumptions are justified.

Fig. 4 of the paper shows a tendency for the mixture to become too lean at high suctions, while it is nearly correct at low suctions. The remedy for this is obviously to increase the size of the main jet,

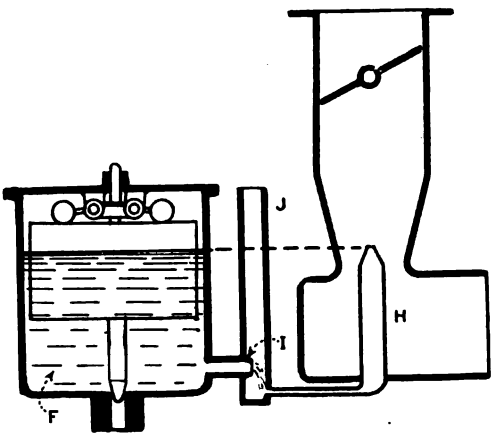

FIG. 24—ILLUSTRATING OPERATING PRINCIPLE OF ZENITH CARBURETER

that is, change the parameter in the main jet equation. For instance, make $F^2 - 4\ (H - 0.2) = 0$. With the same compensating jet at 20 cu. ft. of air per minute, the quality would be 13 oz. gasoline per 100 cu. ft. air. At 100 cu. ft. per minute it would be 9.9, so I must use a smaller compensating jet. I assume one represented by $C = 0.5$, giving only ½ oz. gasoline per minute. The quality varies from 10.5 for 20 cu. ft. to 9.4 for 100 cu. ft. What a difference! The curves in Fig. 23 show the quality curves with $C = 1.0$ and with $C = 0.5$, the flow of fuel through the compensating jet then being respectively 1.0 and 0.5 oz. gasoline per 100 cu. ft. of air.

The assumption has been made that the compensating flow is constant only when the suction is 2 in. of water. It is possible, however, to have constant flow at much lower suctions. This can be made clear by referring to Fig. 24, which shows a constant level fuel chamber F, a well J open to the atmosphere, a fuel orifice I below

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the level, and a connecting channel terminating in this nozzle *H*. The suction at which the flow through the compensating jet *I* becomes constant depends on the areas of *H* and *I* and on the height between *H* and the connecting channel. At exceedingly low suctions, the fuel drops in the well *J*, and a suction is soon reached at which the air passes through the channel from *I* to *H*. From that time on the flow through the compensating orifice is practically constant. The value of this constant-flow suction depends on the instrument. In the Zenith carbureter constant flow is reached at a suction much lower than 2 in. of water. The fair average would be 1.1 in. of water, so that we have there one reason why it is legitimate to continue the quality curve of the main and compensated jets nearer the origin of coordinates.

There is another reason why it can be continued still nearer. It is stated in the paper that "when the throttle is wide-open and the engine is slowed down by additional load, the decreasing suction of the engine permits the well to gradually fill with gasoline, thus decreasing the gravity head in the float-chamber and causing the quantity discharged into the well to decrease." This is a peculiar point, hardly known outside our organization, and I do not wonder that an error has been made. After the suction is reached at which air passes from *I* to *H*, we have no gasoline in the channel, but instead an emulsion, which is much lighter than gasoline. Less suction is required to hold and carry this emulsion up in the well, so that if the critical suction is 1.1 in. of water, going up, it does not follow that the well again fills when the same suction is reached, going down; this kind of hysteresis effect causes the flow from the compensating jet to remain constant for suctions much lower than have evidently been assumed.

To sum up, I have taken the premises under which the theoretical analysis has been made in the paper, and have shown that, by simply changing the parameters, which is equivalent to changing the sizes of the jets, I arrive at results quite different. I question, therefore, whether by changing the jets it would not have been possible to obtain with the actual carbureter entirely different and much more satisfactory results.

FRANK H. BALL:—If the curves were based on theory, it would be debatable whether the same law governs the flow of gasoline and air through fixed orifices, but inasmuch as the flow of both have been repeatedly measured through the range covered in carbureter practice and have always been found to correspond approximately to the law in question, it cannot be attacked without refusing to admit the accuracy of the measurements that have been made.

Mr. Heftler suggests that as an experiment the level might be carried at the overflow point in the discharge nozzle, or even higher. That is just what has been done, and the result absolutely confirms the theory regarding such a condition. When the level was at the overflow point the quality diagram was a straight horizontal line

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through the whole length of the quality zone. When the level was slightly higher than the discharge nozzle, the resulting quality diagram curved upward at the slow-speed end; the diagram curved downward at that end when the level was slightly below the overflow point. This, of course, was a fixed-orifice carbureter and had no air-valve.

A successful valveless carbureter was made on this plan by having the discharge nozzle terminate in a shallow cup that prevented the gasoline from wasting when not in service. When in service an air-stream was made to sweep the gasoline from this cup. With an adjustable gasoline level, a perfect quality diagram could be obtained and the quality was under full control down to zero quantity.

Referring now to Mr. Heftler's discussion of the quality diagram, Fig. 3, it is quite true that a smaller compensating jet and a larger main jet would make the diagram more nearly horizontal between 20 and 100 cu. ft. of air, but they would also make the lean spot at 10 ft. still leaner. If this was corrected by changing the by-pass adjust-

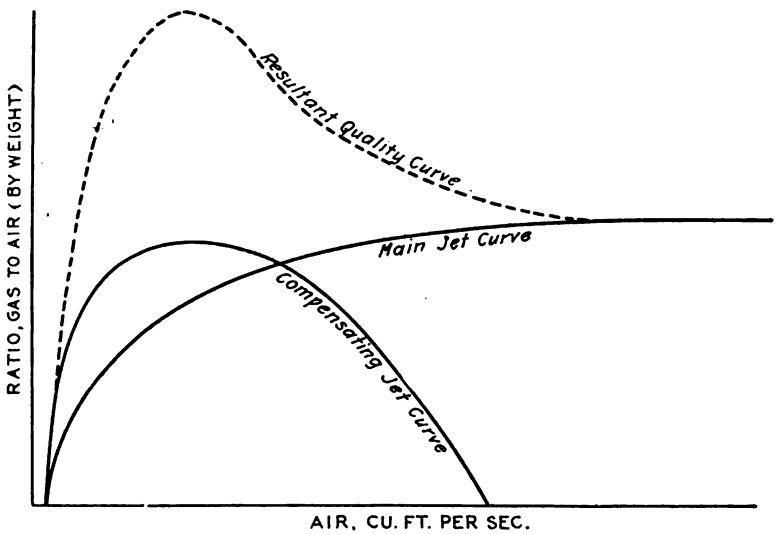

FIG. 25—QUALITY CURVES FOR CARBURETERS WITH TWO VARIABLES IN PARALLEL

ment, the by-pass curve would be moved bodily in a horizontal direction, so that the quality would be too rich for idling. The adjustment from which the diagram in Fig. 16 is made is assumed to be a fair compromise, particularly as it is the setting used regularly by a prominent automobile manufacturer with this type of carbureter.

ACTION OF PLAIN-TUBE INSTRUMENT WITH SIMPLE JET

L. ARNSON:—I would like to call attention to a so-called plain-tube carbureter with a simple jet that reverses the usual action of

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the simple jet mentioned in the paper; I refer to the carbureter developed in the last eight years by George Longuemare, Paris.

The correct quality curve of a carbureter is in the form of a rectangular hyperbola. Mr. Heftler and Mr. Ball in his paper have

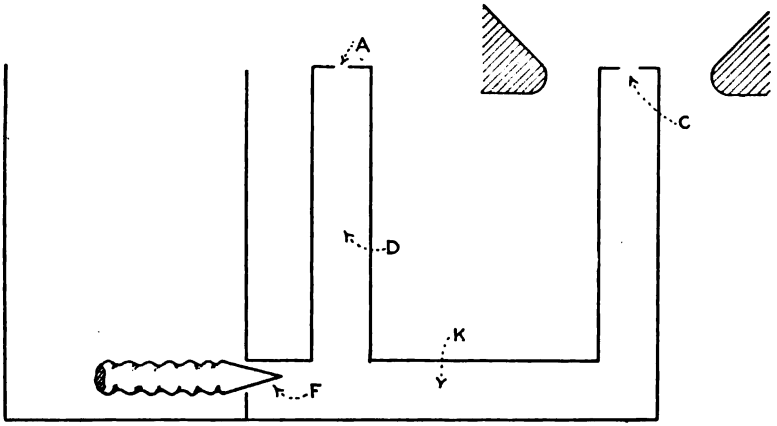

FIG. 26—ILLUSTRATING OPERATING PRINCIPLE OF LONGUEMARE CARBURETER

shown this with two variables (main jet and compensating jet) in parallel, producing the upper curve in Fig. 25. In the Zenith carbureter two variables are in parallel. Longuemare has worked on a different principle in that two variables are in series. This principle is illustrated in Fig. 26. At the end of the jet is a restriction or

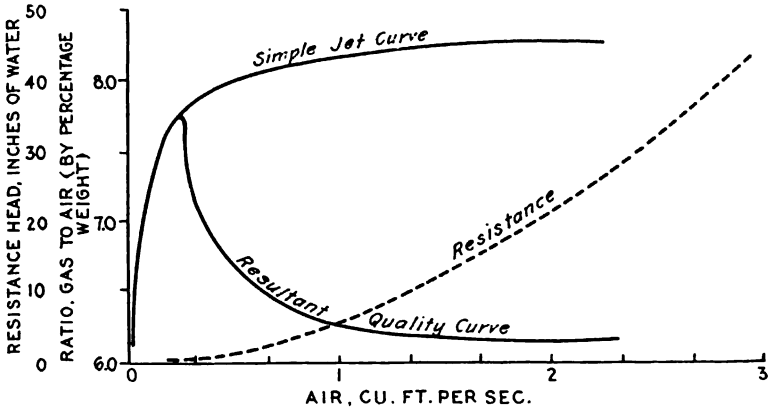

FIG. 27—QUALITY CURVES FOR LONGUEMARE CARBURETER (TWO VARIABLES IN SERIES)

calibrated orifice *C*, and also a fuel orifice *F*, in connection with a large channel *K* and air orifice *A*. The action of this carbureter is that when the suction below begins to act on the jet *F* the chamber *D* empties into the channel *K*. Air bubbles then pass through the chan-

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nel into the jet C. The amount of the infusion of air into the channel is approximately proportional to the speed. The two variables in series are the suction head produced in the channel and the resistance at C, varying with the velocity of the fuel. With increased speed the resistance is increased, because instead of a perfect gas an emulsion lighter than gasoline and heavier than air is passing through K. The resulting velocities are abnormal and do not increase in proportion to the resistance, but at a slightly greater rate.

The equation for the flow of fuel from a jet emptied upon a column of air is

$$V_f^2 = V_a^2 \frac{d_a}{d_f} - 2gh$$

The square of the fuel velocity is equal to the square of the air velocity times the air density divided by the fuel density minus $2gh$, the head effect due to the difference of the jet and fluid level, viscosity, etc. If d_f is reduced as we are reducing it by the emulsion of air, we can get extremely high velocities, and consequently an increased resistance, the tendency of which is to decrease the flow of fuel from the nozzle and at the same time produce an emulsion in the passages. In that way, the mixture is enriched in a simple jet at starting (which is absolutely the reverse of the action of an ordinary simple jet) and becomes leaner with increased velocity. We have, therefore, a simple jet that is the reverse of the jet mentioned by Mr. Ball and that in addition has the necessary characteristics for pick-up without the use of a moving part.

FUEL CONSUMPTION OF TWO-STAGE CARBURETER

The high efficiency limit, Fig. 18, is on normal speeds from about 50 to 140 cu. ft. of air per minute. The air/gas ratio is apparently 15.3 by weight. In Fig. 19 Mr. Ball shows what appears to be a quality curve of a two-stage carbureter. According to this curve when both stages are working the air/gas ratio is 12 at speeds of from 40 to 140 cu. ft. of air per minute. Any design produced in our laboratory showing such a consumption of fuel would be speedily abandoned. We would not permit an instrument to go out at any stage of these speeds with an air/gas ratio less than 15; it must be more than 16.8 with the commercial fuel now available.

FREDERICK O. BALL:—A year or so ago we tested the Longuemare type of compensation. It does compensate, but has the same fault as all fixed-orifice carbureters; the quality falls off to zero at too great a quantity of air. The minimum is not low enough and, as is true of all fixed-orifice carbureters, other means must be used to obtain idling and slow running.

MATHEMATICS UNTRUSTWORTHY

C. P. GRIMES:—We should feel indebted to Mr. Ball for producing a carbureter that, according to his paper, has a working range of

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from 1 to 47. Previous to my present professional connection I spent almost 3 years in what I think to be the finest carbureter research laboratory in the United States, and never found, in testing some 40 carbureters, one with such a range. As a rule they had not half that range.

I have yet to find a case where the truth regarding carburetion is expressed by a mathematical statement. My experiments covered Venturis of various shapes, exits of 8 or 10 different angles, 6 or 8 entrance radii and different exit lengths. I found that each time the Venturi, the exit, or the approach was changed the curve of the air flowing through this Venturi was also changed. I tried jets up to ⅝ in. long and could not get consistent results. As the gasoline comes up in the nozzle and enters the zone of high suction, the more volatile parts of the hydrocarbons are vaporized. When perhaps half way through the nozzle they burst and fly apart and cause a disagreeable hissing noise. This flying apart, I believe, upsets all mathematical theories relating to the flow.

I have obtained curves similar to those shown in Fig. 2 of the paper. These seem to show that the making of a carbureter is a very simple matter; in reality a curve showing the area necessary to maintain a uniform mixture ratio is very irregular. With a jet ⅝ in. long, for instance, the curve will indicate lean mixtures at lower speeds. But if the jet is shortened the curve will be much more uniform. I made some interesting experiments on the height of the gasoline in the carbureter, and found that the depth of the gasoline in the nozzle has a great deal to do with keeping the curve uniform.

I believe that a carbureter giving a little richer mixture at low speeds and a leaner one at high speed is preferable. This point is illustrated in Fig. 3. The intensity of combustion depends upon the closeness of the association of the particles. To obtain uniform explosion pressure a less rich mixture is used at the higher speeds. By compressing fewer molecules closer together, a good power curve is obtained.

FREDERICK O. BALL:—We have endeavored to avoid mathematics as much as possible, because a formula does not mean much unless it corresponds to actual practice. The tests we have made show that in many cases the flow of gasoline and air follow practically the same law. As the fuel gets heavier the curve of flow will be different from the air curve. The different carbureters tested have been taken just as they were used on cars by the manufacturers, or by the individual users.

QUALITY CURVES HAVE SAWTOOTH CONTOUR

F. C. MOCK:—The mathematics and the theory used in connection with carburetion have been founded mostly upon research made years ago with the fuel then common. They are based on the assumption that a uniform mixture of fuel and air gives ideal performance. As

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a matter of fact, the days have gone by, probably never to return, when a uniform mixture can operate an automobile engine under all the temperatures of customary use. If it were possible to take a moving picture record of the mixture proportion, in the same form as the curves in the paper, of one of the best carbureters, the resulting curve would have a sawtooth contour, with variations much greater than any shown by Mr. Ball. These would be necessary in order that the car might give its best performance. This is because the present type of motor-car engine, with ignition by single electric spark, and high speed with consequent short time for flame propagation, will not use efficiently any part of the fuel except that which is in vapor form. The same economic causes that have resulted in the present high price of gasoline have so changed its nature that not all of it evaporates; and further the amount that can evaporate changes continually during the varying conditions of average driving.

TABLE I—PROPERTIES OF AUTOMOBILE-ENGINE FUELS*

Name	Chemical Composition	Density, Deg. Baume	Boiling Point, Deg. Fahr.	Specific Heat	Latent Heat of Evaporation, Cal. per Kg.
Pentane	C_5H_{12}	87	97	0.52	85
Hexane	C_6H_{14}	81	156	0.50	80
Heptane	C_7H_{16}	70	208	0.48	80
Octane	C_8H_{18}	65	257	0.51	81
Nonane	C_9H_{20}	61	302	0.50	.....
Decane	$C_{10}H_{22}$	58	342	0.50	84
Undecane	$C_{11}H_{24}$	55	383	0.50	.....
Tetradecane	$C_{14}H_{30}$	46	485	0.50	.....
		Sp. Gr.			
Benzol	C_6H_6	0.880	176	0.34	93
Ethyl alcohol	C_2H_6O	0.792	173	0.60	220
Methyl alcohol	CH_4O	0.796	150	0.60	284

*Computed values are based on Landolt and Bornstein Physico-Chemical Tables and nternational Annual Physical Tables.

VAPORIZING DEPENDS ON FUEL PROPERTIES

In general we can say: 1. More of the fuel is vaporized at closed than at open throttle; 2. More of the fuel is vaporized when the engine is warm than when it is cold; 3. More of the fuel reaches the cylinders in vapor form when the air velocities are high in the carbureter and in the manifold. The vaporizing conditions change constantly during the operation of a motor car, so that the quality curve

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of a carbureter giving flexibility and power, along with maximum fuel efficiency, must have an irregular contour.

To find the underlying reasons and the laws governing these conditions we must examine the properties of the fuel that concern its use in the engine. These properties, as given in Tables I and II, cover completely, I believe, the requirements of successful engine operation, so far as a fuel supply system is concerned. Further, they explain definitely many peculiar things noticed in engine and carbureter performance.

Petroleum is an irregular natural blend of a number of elements of similar molecular structure, which grade in order as to density, volatility, etc. Table I covers the range of these elements that have been used in gasoline, past and present. The gasoline of the early days was simply the liquid portion of the petroleum that was too volatile to be incorporated safely in the mixture sold as kerosene, and therefore was to a considerable extent a by-product. It consisted mainly of pentane, hexane and heptane. Before the war the fuel used in Europe was also largely of this character. As the demand

TABLE II—PROPERTIES OF AUTOMOBILE-ENGINE FUELS

Name	Chemical Composition	Percentage in Gasoline		Proportion, Vapor to Air, for Perfect Combustion		Critical Temp., Deg. F., of Vapor Saturation			Initial Temp.* Deg. F. of Air to Vaporise Fuel Charge
		Present (1915)	Early	By Weights	By Volumes	Full Cyl. Charge	¾ Cyl. Charge	½ Cyl. Charge	
Pentane	C_5H_{12}	....	10	0.065	0.0272	0	0	0	....
Hexane	C_6H_{14}	3	45	0.066	0.0220	0	0	0	....
Heptane	C_7H_{16}	12	35	0.066	....	....	....	....	....
Octane	C_8H_{18}	30	10	0.066	0.0170	68	54	33	98
Nonane	C_9H_{20}	35	....	....	....	....	....	....	....
Decane†	$C_{10}H_{22}$	15	....	0.066	0.0136	109	95	72	152
Undecane	$C_{11}H_{24}$	5	....	....	....	....	....	....	....
Benzol	C_6H_6	....	....	0.075	0.028	61	59	56	114
Ethyl alcohol	C_2H_6O	....	....	0.111	0.070	57	41	24	257
Methyl alcohol	CH_4O	....	....	0.155	0.140	65	52	36	384

*Initial temperature of gasoline 90 deg. F.

†Loss of density of air charge for complete vaporisation equals 10 per cent (90 deg. F. equals 100 per cent).

for motor-car fuel increased it was necessary to use, year by year, more and more of the petroleum elements, with the result that to-day gasoline runs from octane to undecane and contains scarcely a trace of its earlier ingredients. Tetradecane is an element typical of what is now sold as kerosene; its molecule is heavy and complex.

A molecule of tetradecane can be broken to form one molecule of hexane and one of heptane, with one atom of carbon left over. This is practically what is done by the Burton and Rittman cracking processes, which, under the combination of high temperature and high

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pressure, with no oxygen present, turn the heavier petroleum elements into lighter ones, with a carbon residue. This process can take place in cylinders, when we try to run the engine cold with a rich and largely unvaporized mixture and is often the cause of the troublesome carbon deposits on cylinder walls and spark-plugs.

In Table II are shown some properties of the fuel elements that concern more closely their adaptation to use in the engine. The percentage of the different elements varies throughout the country, but those given are typical of the fuel available in the central and eastern States. The early gasoline was almost entirely different in composition from that of the present day. European gasoline (petrol) is similar to our early gasoline, but contains more octane.

The proportions *by weight* of fuel vapor to air for perfect chemical combustion vary but little, as the actual quantities of hydrocarbon must be the same in all cases. The proportion *by volume* of vapor to air, however, decreases markedly as we come to the heavier gasoline elements. This is because the molecules are heavier and therefore fewer are needed to make a given weight; according to an elementary law of physics equal volumes of gases (at the same temperature) contain equal numbers of molecules, so that for the same weight of the heavier elements the volumes must be less as the molecules are heavier.

A recent bulletin of the Bureau of Mines on Inflammability of Mixtures of Gasoline Vapor and Air indicates that the limits of inflammability with an electric spark depend more upon the ratio of volumes than upon the ratio of weights and that the lower limit is around 0.014. This limit would exclude decane and undecane, which are important components in practically all the gasoline sold to-day. It was found, of course, that under the heat of compression the limits of inflammability were extended. This simply emphasizes the fact we know from experience, that the proportion of fuel needed is based on the amount vaporized in the manifold rather than upon the amount fed from the carbureter.

ACTION OF CRITICAL TEMPERATURES

The most important part of these tables is that giving the critical temperature necessary to allow the existence of enough vapor for a running mixture. The maximum density of the vapor of any liquid is determined and limited by the temperature. This density is called the saturation point and beyond it no more vapor can be gotten into a given space; any compression will result in the condensation of enough of the vapor so that the density is the same as before. Thus each temperature gives a different density of saturation; conversely, there is a different saturation temperature for each density.

What has happened to engine fuel is this: A point has been reached where at ordinary operating temperatures the vapors of some of the elements of gasoline cannot be made dense enough to furnish

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their full share of the running mixture. The proportion of the running mixture that any element can furnish in vapor form depends on the temperature and the density of charge required. For instance, referring to octane, to furnish a vapor sufficiently dense to carburet a full charge requires a vapor or mixture temperature of 68 deg. F. If the throttle is partly closed, however, so that only two-thirds of an air charge is taken in, of course only two-thirds of the vapor charge is needed, or two-thirds the density, which can then be obtained at 54 deg. F. At one-third of a cylinder-charge, which corresponds to a speed of 20 m.p.h. on a smooth city street, the necessary vapor can be obtained at 33 deg. F.

A certain drop in temperature takes place while the gasoline changes from a liquid to a vapor state. This is usually compensated for by preheating the air. With octane the air charge must be preheated 30 deg. F. What actually happens under driving conditions is this: Driving at 20 m.p.h. (closed-throttle or nearly closed-throttle position) the 30-per cent octane in the gasoline becomes vaporized if the air taken in is at 63 (33 + 30) deg. F. But if the driver opens the throttle to accelerate, although more gasoline will be admitted through the carbureter, no more octane can be vaporized at the temperature, and it requires instead 98 (68 + 30) deg. temperature before it will become vaporized in sufficient quantity to give the full (wide-open) throttle mixture.

For decane much higher temperatures are required. The mixture temperature for wide-open throttle is 109 deg. F. The drop of temperature for vaporization is 43 deg. F., so that a wide-open throttle charge requires an initial air temperature of 152 deg, or a closed-throttle temperature of 115 deg. In many cars the air-intake temperature during the eight colder months of the year is not much over 120 deg. As a result the drivers find that although they can run on an economical setting, that is, using all the components of the gasoline, at closed throttle, at wide-open throttle the engine will not fire. The gasoline orifice must then be opened so that at wide-open throttle enough fuel for a firing mixture is obtained from the heptane, octane and part of the nonane elements of the gasoline. The remainder of the nonane, decane and undecane, although fed from the carbureter, is in the form of liquid or drops in the intake manifold and is practically wasted.

These are the reasons for what is generally observed in practice, that we have to use a richer mixture for wide-open than for closed-throttle positions, and why wide-open throttle operation is much more sensitive to temperature than closed-throttle operation.

OPERATING DIFFICULTIES WITH HEAVY FUELS

There are other disadvantages in using heavier fuels. Preheating the air involves expanding it and decreasing its density, with the result that for a given suction less air charge is taken into the

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engine. Even if the necessary vapor temperature be maintained without preheating the air, there will be a certain loss in density as long as the air is taken in along with and through the same passage as the gasoline vapor. In the case of decane the loss from preheating the air amounts to 10 per cent, which shows that we have, even with our present fuel, reached a point where to get flexible operation (with a uniform-mixture carbureter at least) it is necessary to sacrifice volumetric efficiency. With kerosene the mixture temperature is so high that this loss in charge volume is serious indeed, making necessary larger piston displacements for the same horsepower output. I believe that this fact alone, exclusive of the difficulty of maintaining a sufficiently high temperature, makes it impossible to use kerosene below tetradecane in motor cars of the present type.

Tables I and II show the fuel properties for benzol, ethyl (grain or sugar beet), alcohol, and methyl or wood alcohol. Benzol grades in quality slightly better than octane and would be a good substitute for gasoline for motor-car use, if it could be obtained. It has the disadvantage that it freezes.

The alcohols will run on low mixture-temperatures, but their latent heat of vaporization is high. This, in conjunction with the fact that more than twice as much alcohol as gasoline is needed for a running mixture, makes it necessary to preheat the air to a high temperature; in fact, to a point hardly obtainable in a motor car. Even an exhaust-jacketed intake-manifold will not furnish sufficient heat under average closed-throttle driving. A high thermal efficiency is obtainable with alcohol on account of the fact that high compressions can be used with a corresponding increase in the efficiency of the heat cycle. At all times the volumetric proportion required of fuel to air is more than double that of the gasoline hydrocarbons, and the consumption per horsepower hour is always greater.

C. P. GRIMES:—At present engine manufacturers are confronted with a serious problem, caused by some carbureting devices. It has been found that after putting clear lubricating oil in the engines and running them for several hundred miles, perhaps 10 per cent of the fluid in the crankcase is gasoline or kerosene. Mr. Mock explained this action. The heat is not sufficient to fracture the fuel or at least to maintain it as a vapor.

ATOMIZING AND VAPORIZING FUELS

E. H. SHERBONDY:—We ordinarily think of a carbureter as a device separate from the complete engine. As a matter of fact, the intake manifold and what we call the carbureter serve separate functions in preparing the combustible mixture for use in the engine cylinder. Fundamentally, all the carbureters in this day of heavy fuels act only as weighing devices, giving a certain weight of fuel to a certain weight of air. This fuel must be atomized. The atomization requires a certain amount of energy. In almost all carbureters

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the arrangement for atomizing is very poor. A butterfly throttle-valve is placed immediately above the fuel nozzle and is partly closed. The fuel is thereby thrown out of the mixture, resulting in bad distribution.

Following atomization, we attempt to vaporize. Some vaporization starts with light fuel immediately at the fuel nozzle, but about 80 per cent of the fuel that goes into the cylinders is vaporized not in the manifolds but in the engine cylinders.

The engines as designed to-day are arranged so that the fuel has to be carried up through the valves. A system arranged so that the fuel travels from the fuel nozzle through the intake manifold and valves in a downward direction would assist the travel of the fuel greatly. Then if the engine itself were arranged to perform the function of atomization, the fuel would be ready, after passing into the cylinder, to be vaporized by the heat of compression. In the Diesel engine, which does not require an ignition device, the fuel is introduced by spraying under high pressure into the cylinder and is immediately vaporized.

VAPORIZATION INSTANTANEOUS

Vaporization is often thought of as requiring a relatively long time to take place. For instance, if we take a square foot of surface and wet it with gasoline, we observe that the gasoline apparently vaporizes slowly until it is all gone. Vaporization appears then to require a time interval, but this is not the case. The particles on the top of the surface change spontaneously from a liquid to a vapor condition. Absolute proof that vaporization is instantaneous is found in the indicator card of the Diesel engine. If any period were required for vaporization, the curve would drop.

I think that the future of the automobile industry hinges on the development not of carbureters but engines. So far a weighing device has been developed, but the carbureter engineer has dealt with only a small portion of the problem of carburetion. He has been greatly handicapped, largely by the construction of the engine.

LAW OF FLOW THROUGH AIR-VALVE

GEO. W. SMITH, JR:—Fig. 8 of Mr. Ball's paper shows that the air supplied by an air-valve weighted with a spring is directly proportional to the head. The area of the opening varies directly as the head and the velocity varies directly as the square root of the head. Therefore, the flow should be represented by a curve instead of by a straight line. According to my explanation, the flow of air varies as the three-halves power.

FREDERICK O. BALL:—We have obtained with certain carbureters a curve of air that was made to coincide with the curve of gasoline, the two being identical throughout the full range. We have also succeeded in getting an annulus with enough friction that the curve of the flow of gasoline is almost a straight line.

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ANDREW AUBLE:—On one particular economy run a normal mileage was obtained on the first half gallon of gasoline, and on the next or last half gallon more than double the mileage was obtained. I believe this was due to the fact that the engine became hot enough to evaporate the vapor before the explosion took place in the cylinder. When the engine was heated to a point where I felt it was receiving fuel in an evaporated form, it was much more efficient, the economy being brought to a point hardly thought possible. When the gas reaches a high temperature the engine temperature decreases automatically. My experience with an air-cooled engine was that it would become heated to a certain point, and then with continuous running eventually become cooler and remain so in mile after mile of low-gear testing. I believe this was due to the fuel being evaporated instead of vaporized.

FUEL TRAVEL IN TWO-STAGE CARBURETER

E. H. SHERBONDY:—The device shown in Fig. 20 of Mr. Ball's paper has a gasoline passage from the float-chamber feeding to the two nozzles. When operating normally under low throttle, the fuel would be discharged through nozzle *3*. For fuel to be discharged from that nozzle compression must take place in the gasoline passage, thus tending to throw the fuel through the second nozzle, even though the throttle valve *7* be closed.

FREDERICK O. BALL:—The gasoline level is lower in the second nozzle than in the float-chamber, owing to the resistance of the passage between the primary nozzle and the float-chamber. If this resistance were not kept large, all the gasoline would be drawn from the second-stage nozzle and air would go in that way, destroying the action of the primary carbureter.

We recently made a demonstration of the flow of gasoline through an orifice and a friction annulus. The apparatus used consisted of a chamber with a float to maintain a constant level. The chamber was mounted so that it could be raised or lowered, thus changing the head on two discharge openings. The discharge openings were at the same level and connected with the float-chamber. One discharge opening had a fixed orifice and the other a friction annulus similar to that shown in Fig. 12.

The float-chamber was located so that the gasoline level was about 1 in. above the discharge nozzles. The amount of gasoline flowing through both nozzles was then adjusted so that the flow was equal as measured by graduates arranged to catch the discharge. Next the float-chamber was raised so as to make the head about 20 in. If the same law of flow applied the discharge from both openings would be equal at the higher head, but the discharge from the friction annulus was really more than twice that from the orifice.

The amounts of gasoline discharged at 1 in. head and 20 in. head are in the same ratio as the air the carbureter must deliver to keep

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the mixture correct. If two carbureters, one with a regular orifice and the other with the friction annulus, have the same capacity at 1 in. suction in the mixing chamber, then at 20 in. suction, since the friction annulus delivers twice as much gasoline as the orifice, it can also deliver twice as much air; in other words, the friction-annulus carbureter covers a wide range of capacity with small range of suction compared to the orifice carbureter.

SERVICE CONDITIONS NOT CONSIDERED

P. S. TICE:—The authors are to be congratulated on the ingenious graphic method employed to present the outstanding characteristics of compensation in the several well known types of carbureters. The diagrams, however, have seemingly been constructed largely from data obtained under conditions materially different from those obtaining in service on engines. Therefore the diagrams are apt to be misleading. It is regrettable that more space was not devoted to a description of the apparatus and a summary of the methods employed in its use. A case in point is presented by Figs. 1 and 2 of the paper and the statements that accompany them. It is assumed apparently that the flows of air and fuel are caused by identical heads or differences of pressure. In the type of carbureter (elemental fixed-orifice) here considered, this condition is impossible of realization. Not only is the mean head causing efflux of liquid always greater than the corresponding mean head causing the flow of air, but also the two do not bear a fixed ratio, in that the head upon the fuel orifice increases at a higher rate than does that impelling the air flow. A further distinction arises in a consideration of the relative quantities of each fluid delivered under given mean heads in that the densities of the two are widely separated, and the discharges do not therefore bear the same relationship under different conditions of installation, even though the mean heads remain identical in value or relationship. Thus it follows that if any carbureter, of any type, be set or fitted to any engine, an investigation of that carbureter when removed from the conditions of its installation on that particular engine will be valueless as a statement of its characteristics, even for purposes of comparison with other carbureters investigated with it under like experimental conditions.

A practical carbureter is not merely one of its given nominal size with hard and fast characteristics displayed under any and all conditions; but it is a Packard carbureter, or a Ford carbureter, or a Hupmobile carbureter, and will display its true characteristics only upon investigation conducted on its home lot, as it were. If the depressions and velocities induced in the carbureter were constant at a given engine speed, the comparison of characteristics would be simple and not involve the engine and its type. Then and then only could the admirably simple and controllable ejector be made to give usable readings of heads and quantities.

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DISCHARGE THROUGH ORIFICES

In those portions of the paper dealing with the flow of fuel through an orifice reference is had to the factors acceleration, inertia and friction. In carbureter practice it is insufficient to call the fuel passage an orifice, simply, and to assume that the discharge from it will or can vary directly as the square root of the head. With the sizes of fuel orifice permissible in common carbureter practice, it is, practically speaking, a physical impossibility even to approach a discharge directly proportional to the square root of the head. With any permissible orifice having length, that is other than a true sharp edge without approach, the coefficient in the expression $V = C\sqrt{2gh}$ increases with the head, the rate of increase being controlled in a general way by the ratio of the length to the diameter of the orifice.

I am thoroughly in accord with the authors in the matter of the importance of atomization of the fuel. It is safe to say that it is impossible, under the conditions and with the values to be worked with in carburetion, to carry this phase of the carbureter's performances too far—provided, of course, that its other functions are not curtailed.

The atomizing at other than full throttle, or before the second-stage butterfly valve is well opened, can be assisted by the air drawn through nozzle *6* and discharged with fuel through nozzle *3*. This action would take place unless the common fuel passage to both nozzles were extremely large and the outlet orifice of nozzle *3* correspondingly small. The pick-up device is clever, susceptible of control in fitting, and should be of great value with certain types of engines.

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FACTORS IN UNIVERSAL JOINT DESIGN

By A. Ludlow Clayden

(Member of the Society.)

Abstract

The author considers the effects of velocity variation on the operation of a car and states that this variation is absorbed mainly by the flywheel. A formula is given for calculating the pressure on universal bearings. Various methods of protecting and lubricating joints are described.

A number of European types of joints are illustrated. A much larger number of types of joints are used abroad inasmuch as each maker usually makes his own design instead of purchasing it from a specialist as is the usual practice in this country. In conclusion the paper describes types of joints using flexible material, such as leather or spring steel.

Hardly a part in an automobile has more work to do and yet has less attention than a universal joint. It is liable to neglect, alike from the user of the car and the engineer responsible for the chassis layout. Usually made by specialists, the quality of material and workmanship in the average universal are of a high order of merit, but more attention to the disposition and to means for maintaining lubrication would result in higher chassis efficiency and greatly enhanced durability. The way in which the user has become accustomed to poor durability of universals can be noticed by appreciating one's own natural expectation that a car that has run 10,000 miles will have a good deal of slack and rattle in the transmission.

When properly proportioned to the work it has to do, and when properly lubricated, few parts of a machine are more efficient

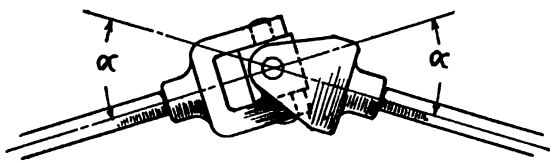

Fig. 1—Diagram to Illustrate Velocity-Ratio Variation

than a universal joint. This was shown by demonstration in tests made on a well-known joint in the laboratory of the mechanical engineering department of the University of Kansas. The most important result of the tests was that the efficiency of the joints was close to 99 per cent for angular deflections up to 7 or 8 deg. It was also shown that the efficiency fell off more and more rapidly as the angle increased, but unfortunately the largest angle for which determinations were made was about 15 deg. In any case, however, the loss of power in a well lubricated universal is so small that it can be neglected.

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EFFECTS OF VELOCITY VARIATION

Where a single universal is used to connect two shafts at an angle, if the first shaft runs at a steady and regular speed, the second shaft will run at a speed that varies during each revolution between maxima and minima, the values of these depending upon the angle of deflection. The change in the velocity of the driven shaft is rapid, and as it occurs during each quarter revolution, there are two maxima and two minima during each complete turn of the shaft. The formulas for determining the amount of variation are

$$V_1 = V \cos \alpha \text{ and } V_2 = \frac{V}{\cos \alpha}$$

where V = angular velocity of the driving shaft, V_1 and V_2 = the maximum and minimum angular velocities of the driven shaft, and α = the angle of intersection of the produced axes of the shafts. (See Fig. 1.)

Variation in velocity of a propeller-shaft in a car having one universal only cannot take place without some curious reactions. At one end we have the inertia of the whole car as a moving mass tending to keep the road wheel turning at a constant velocity, and at the other end there is the flywheel of the engine. Obviously we cannot accelerate and retard the road wheel, and so the whole car, twice during every revolution of the propeller-shaft, even by a small amount, and it is to be assumed that the variation, which must take place somewhere, does so mainly on the flywheel.

FLYWHEEL FLUCTUATIONS

Owing to the power of an internal combustion engine being made up from a series of explosions the flywheel has natural periodic fluctuations of its own. These will vary with the number of cylinders, the speed of the engine, the weight of the flywheel and the force of each explosion. With a four-cylinder engine the fluctuations take place at the same rate as those due to the universal; in an eight at double that speed; so a four may have flywheel fluctuations that either coincide with or neutralize those due to the universal joint. That these fluctuations in speed are not so small that they can be neglected can be shown by quoting an actual value and, to take a low speed where variations have the greatest chance of being perceptible to the passengers in the car, we can assume 500 r.p.m. with an angular deflection of 10 deg. At 500 r.p.m. there will be 1000 maxima and 1000 minima of velocity in the propeller-shaft, which means 1000 vibrations per minute or about 17 per second. At the maximum positions the driving effort on the teeth of the bevels will be heavy, and at the minimum it will be light. An alteration in tooth pressure at so slow a rate of change as 17 per second is distinctly perceptible in the form of vibration.

With a steady speed of the driving shaft of 500 r.p.m. and an

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angle of joint of 10 deg., the maximum velocity of the driven shaft would be 508 and the minimum 492 r.p.m., equal to a variation of 16 r.p.m. each quarter revolution. Now a quarter revolution at 500 r.p.m. takes 0.03 sec. An acceleration at the rate of 16 r.p.m. in this short space of time being rapid, the load imposed on the bevel teeth is large. Practice shows clearly enough that cars with one universal joint are satisfactory, so the velocity variation is not serious. But it is probable that the elimination of the variation would render very smooth and effortless low-speed running on high gear somewhat easier of attainment.

WAYS OF ELIMINATING VELOCITY VARIATIONS

There are two ways of eliminating the velocity variation, and both have been used successfully in practice. If we use two joints, one at each end of the propeller-shaft, and if these make equal angles,

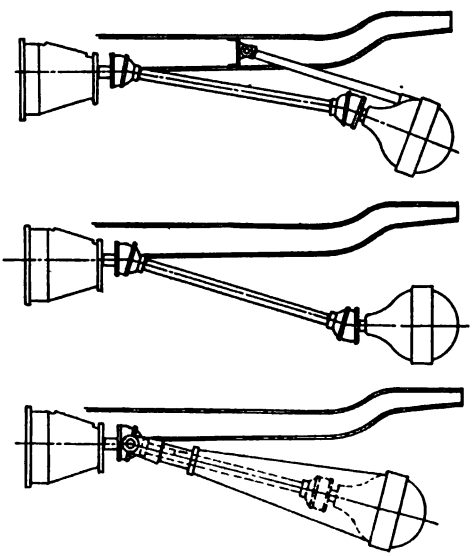

FIG. 2—THREE TYPES OF PROPELLER-SHAFT LAYOUT
Upper—Recommended by C. W. Spicer to Divide Angularity Evenly Between the Two Joints. Center—Obtained with Hotchkiss Drive, Bending Both Joints Equally. Lower—Long Torque-Rod, Rendering Rear Joint Inoperative.

the propeller-shaft will vary in velocity owing to the angularity of the front joint, but the back joint will neutralize this and the bevel-pinion shaft will run as steadily as the flywheel, any fluctuations of the speed of the crankshaft being reproduced faithfully on the pinion without reference to the universals.

The angle between the gear-shaft to which the front universal is attached and the propeller-shaft must be equal to that between the propeller-shaft and the bevel-pinion shaft. This can be arranged either by keeping the two shafts always parallel or by dividing the

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angular displacement between the two universals equally. If we adopt the latter we gain the advantage that the angular displacement of either joint is only half what it would be otherwise, and the efficiency is therefore better by some small amount. It is also unquestionable that the divided-angle design is simpler to carry out than the parallel-motion scheme, as the former can be secured by the use of a single simple torque-rod, while the latter requires regular parallel-motion linkage of some complication.

It may be pointed out, however, that the so-called Hotchkiss drive, where the axle is locked rigidly to the springs, keeps the bevel shaft very nearly parallel to the ground at all times, and so it is perhaps simpler, from this viewpoint, to use the parallel arrangement than the divided angle. The emphatic point is that the usual sort of torque-rod anchored to the frame close to the front universal is not good, as it merely prevents the rear joint from doing its fair share of work, and the second joint from damping out the velocity variation, lowering the efficiency by throwing all the angular deflection upon the forward joint. Fig. 2 shows the three designs.

However, the efficiency of joints is high and the velocity variations are of fairly small importance compared with the difficulty of lubricating the universals properly and their liability to wear and rattle. Proper layout will minimize wear, but it does not help lubrication or prevent the creation of the slackness that produces noise.

Originally, universals were crude affairs, were unprotected and were unprovided with any means for lubrication, but of late years some elaborate designs have been worked out, and in a few cars the universals can be trusted to last as long as any other part. The secret of durability is twofold: first, the universal must be large enough for the load; and, secondly, it must be bathed in lubricant continuously.

If the bearing surfaces are too small the torque of the engine will be able to squeeze out the lubricant; and it must not be forgotten that the torque of modern engines is very high indeed. Calculation will show that among cars of all nations the bearing pressures in the universals when torque is at its maximum can vary from a few hundred to several thousand pounds.

CALCULATION OF BEARING PRESSURE AND JOINT SIZE

The pressure on the bearings of a universal can be found by the formula

$$P = \frac{Hp \times 33{,}000 \times 12}{D \times \pi \times A \times R}$$

Where P = pressure in pounds per square inch, D = effective diameter of joint in inches, A = area in square inches of the two bearings carried by one fork, and R = the minimum number of revolutions per minute of the shaft at which the horsepower (Hp) in question can be transmitted.

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The pressure ought not, in the opinion of many engineers, to be allowed to exceed 1000 lb. per sq. in. at the maximum torque the engine is able to apply on low gear. However, 2000 lb. is often permitted and even double this pressure, but the durability of the joints naturally suffers. A joint large enough to keep the maximum pressure below 1000 lb. per sq. in. is not cumbersomely big for passenger-car use; it costs more than a smaller joint, but the extra outlay is well recouped in the better durability.

Just to consider an extreme case assume an engine developing 70 b. hp. at 2100 r.p.m. If the low gear in the gearset is 3 to 1, we get

$$12{,}600 = P \times D \times A$$

Or, if P is 1000 lb. per sq. in.

$$D \times A = 12.6.$$

Thus if there are two pins each with an area of 1 sq. in., the mean diameter of the joint should be 6 in.; or 3 in. with pins twice the size.

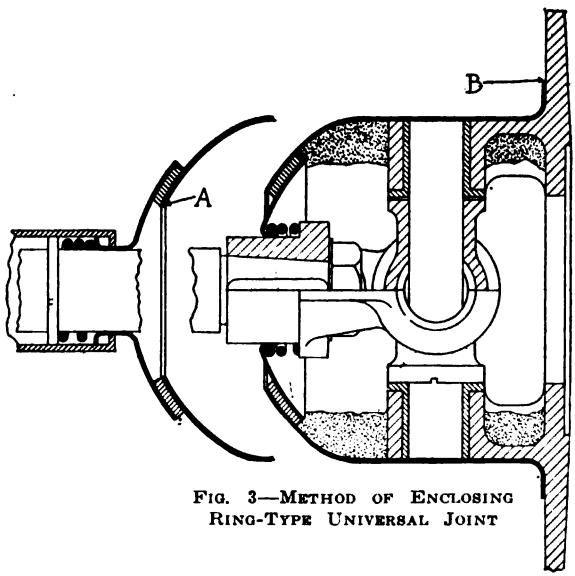

FIG. 3—METHOD OF ENCLOSING RING-TYPE UNIVERSAL JOINT

These figures, rough as they may be, serve to show that the size of joint necessary to give a safe bearing pressure is not excessive.

It would be impossible to do more than guess at the safe maximum load, but that the lower it is the longer the life of the joint goes without saying; also the poorer the system of lubrication the lower should be the pressure. The fact that the joint does not revolve on its bearings, but merely oscillates slightly, makes lubrication more difficult, and this fact must be allowed for.

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A convenient formula for determining the safe diameter for a universal, when the size of the pin bearings and the maximum pressure have been chosen, is

$$D = \frac{Hp}{R \times A} \times \frac{126{,}000}{P}$$

The same formula can, of course, be used to determine A when D is fixed by other considerations.

In America such good universal joints have been made by specialists that automobile engineers have given little attention to their

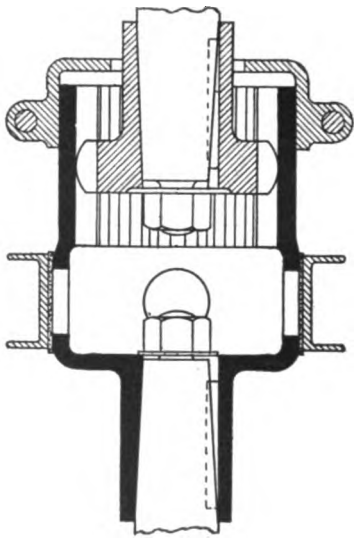

Fig. 4—Gear Pinion Telescopic Joint Used on British Commercial Cars

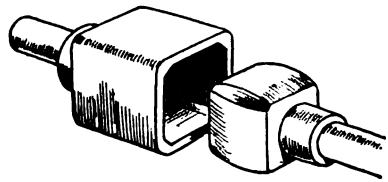

Fig. 5—Combined Universal and Sliding Joint; Used on British Lanchester Car

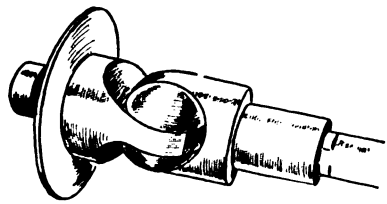

Fig. 6—Ball-Shaped Universal Joint; Used on British Rover Cars

FIGS. 4 TO 6—UNIVERSAL JOINTS USED ON BRITISH AUTOMOBILES

design, but in Europe each automobile manufacturer makes his own joint, as this is a part of the chassis never taken up by specialists with the solitary exception of the Timken representatives, who have shown some remarkable durability with a taper roller-bearing construction. As a result, the European makers have been more concerned to protect and lubricate the universals properly, having nobody else upon whom to place the responsibility.

PROTECTION AND LUBRICATION OF JOINTS

When the idea first arose that a universal needed protection from mud, covers were made of leather. These were of little use, as the centrifugal force drove the grease through every pore, straps broke and mud had a rotting action, so that the cover was soon destroyed. The

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next step was to use a pressed brass cover and to close the mouth of the case with a smaller leather cover which, in this case, was just a plain cylinder. From this point development led to the use of a cover pressed or spun into a hemispherical form, the opening necessary to allow free movement to the shaft being closed by a corresponding pressed hemisphere held up by a coil spring outside the shaft. The last-named type of covering is in greatest use to-day, and is very satisfactory if made properly.

All the different forms of case were supposed to be packed with grease, and grease is still the most common lubricant for universals. A well designed joint will contain enough grease to suffice for from 3000 to 5000 miles of running under ordinary conditions. There is one

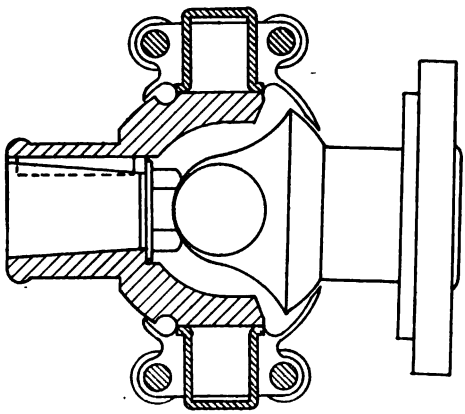

Fig. 7—Ring Joint Used on Wolseley Car

particular point to be kept in mind here, the effect of centrifugal action on the lubricant. If the outer case is large the grease can all be thrown to the ends and the universal then runs in a cavity at the center where no grease touches it. Thus the case ought to fit the joint fairly closely. Next, the diameter of the case should be sufficiently greater than that of the hole through which the shaft protrudes, to allow for the retention of a fair quantity of grease, supposing the piece that closes the hole to be removed.

If a joint casing is filled completely, centrifugal action will cause the middle part of the grease to be forced out through the joint between the case and the cover cap, however good the packing gland may be; the action is slow but sure. Thus the joint and case must be so proportioned and so arranged that the pins and their bushes are as close as possible to the outer circumference of the case and can be lubricated perfectly without the retention of grease at the very middle of the joint. In Fig. 8 is shown where the grease can be reasonably

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expected to remain. Even with these precautions in design the user should be careful not to use grease that is very hard, because a certain amount of fluidity is needed to allow the lubricant to work its way into the pins.

Looking at Fig. 3 there might be reason to suppose that oil could be used, as there is no way for it to escape when the car is standing. However, most attempts to use oil have been disappointing because of the splashing that takes place. The usual way of arranging the cap is as shown at the left in Fig 3. Presumably the oil is thrown against the inner surface of the cap at *A*, whence it is ejected past the packing. It seems thus that there are theoretical advantages to be claimed for inverting the usual arrangement and fitting the cap as shown on the right in Fig. 3, but even so it is to be doubted if oil would remain in the cover long, and there would have to be some sort of a joint at

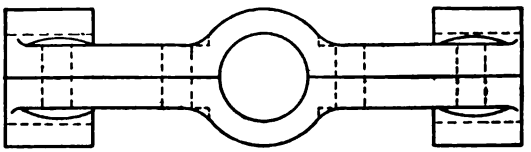

FIG. 8—RING JOINT TYPICAL OF EUROPEAN DESIGN

B to allow the case to be assembled. This design has been used with success on several cars.

STATIONARY CASES TRIED

Experiments have been made with cases entirely independent of the joint itself and not revolving with it. A case was arranged to be supplied with oil from the gearset, the mouth of the case being closed by a leather tube having a slip-ring with free fit on the shaft. The device worked well, but it was clumsy, heavy and awkward in assembly. There was also some trouble to insure sufficient oil without permitting the passage of too much. A modification that resulted in considerable improvement was to abandon the oil-feed from the gearbox and substitute a copper pipe leading to an oil-cup on the side of the chassis frame. This provided an easy way to charge the case with oil, but the user of the car still had no way of telling whether he was putting in enough till it began to overflow.

DESCRIPTION OF EUROPEAN TYPES

One of the most difficult types of universal to make so that it will be satisfactory under all conditions is that which includes a sliding joint. Probably the originator (at least as far as automobile practice is concerned) of the combined universal and telescopic coupling was the DeDion Company, which was compelled to use a

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joint of this nature in connection with its axle. The old DeDion cars had the transmission and differential in a unit attached to the rear of the frame. The wheels were carried on a tubular axle that was slightly dropped, and the drive from the sprung differential to the wheels was made by a pair of short propeller-shafts with universal joints at each end. For this purpose the familiar type of cylindrical outer casing was used, this having two deep grooves internally. The shaft had a cross-pin on which were fitted square nuts of phosphor bronze, free to turn on the pin and with the tops rounded. There is a fundamental difficulty in lubricating this type of joint with grease, this being the pumping action which tends to pull the lubricant outward. The difficulty can be overcome by careful inclosure and a great many ingenious devices have been evolved with this end in view.

Another drawback to a joint of this design is the double wearing surface. There is opportunity for looseness to develop between the

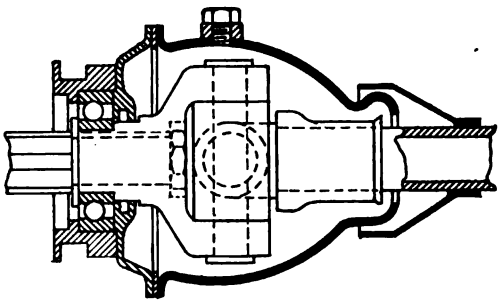

FIG. 9—STATIONARY COVER USED ON BRITISH CHASSIS

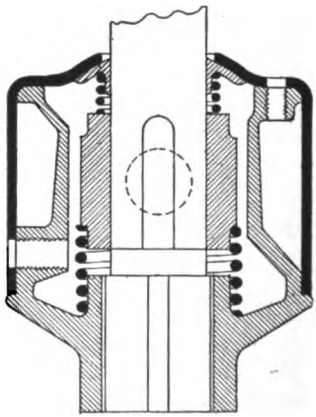

FIG. 10—METHOD FOR ENCLOSING JOINT OF DE DION PATTERN

sliding nuts and the outer case, while wear also takes place on the pin and on the bore of the nut that bears against the pin. This gives no trouble, however, if the parts are made of the right material and if the mounting is such that the telescopic action is not excessive.

An alternative design that is used considerably in America and abroad is the spline coupling, one end of the propeller-shaft being splined and the fork of the universal joint at that end grooved to correspond. This makes a good telescopic joint if there is ample length of bearing, but, owing to the necessarily small radius, the pressure on the surface is high. If ample length is not provided, it will be impossible to retain grease between the splines, and as soon as the lubricant is squeezed out, wear will be rapid.

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APPLICATION WITH HOTCHKISS DRIVE

At the present time there seems to be every reason to believe that the Hotchkiss drive will shortly become conventional practice and attention may be drawn to the fact that a rear spring that is normally nearly flat provides the ideal form for Hotchkiss drive from the viewpoint of universal joints; since with a flat spring the fore-and-aft movement of the rear axle will be at a minimum. There are some designs in which the amount of slide on the universal joint from one extreme of spring motion to the other is less than half an inch. This

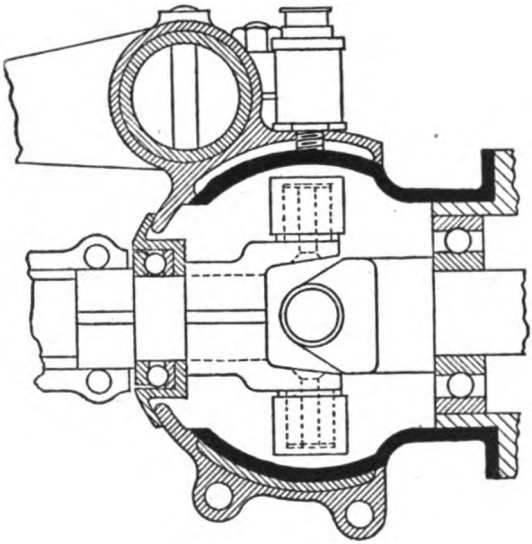

FIG. 11—TYPICAL BALL-JOINT ENCLOSURE AT FRONT END OF TORQUE TUBE

means that in ordinary running the movement is so slight as to be almost negligible.

A rather curious type of telescopic joint, Fig. 4, has been used considerably on British commercial vehicles, being more durable than the DeDion type of sliding joint. This consists of a small pinion on the end of the propeller-shaft, having the tops of the teeth cut to a curve with the center of the pinion as radius. The teeth slide in grooves cut in a cylinder case which is attached to the final drive pinion just like the outer part of the DeDion joint. It is claimed that this design allows a larger area with less weight, and it also eliminates the necessity for the cross-pin used to carry the nuts in the DeDion joint.

A very different type of combined universal and sliding joint that is still used on the Lanchester car is the simplest of all. In this, Fig.

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5, the end of the propeller-shaft carries a block of steel about 3 in. square. The block is about 1¼ in. wide and the edges are all curved. On the gear-shaft is a steel cup of square section into which the block on the end of the propeller-shaft fits snugly. Unless carefully made it is almost impossible to eliminate rattle from this type but as made by the Lanchester Company it has been successful and adhered to by that concern for nearly ten years.

There have been many unusual universal joints, some of which possess special merit. One, used by several British firms, notably the Rover Company, is illustrated in Fig. 6. There are two flat forks, connection between them being made by a steel ball with two grooves cut in it at right angles. This same joint was used in one instance

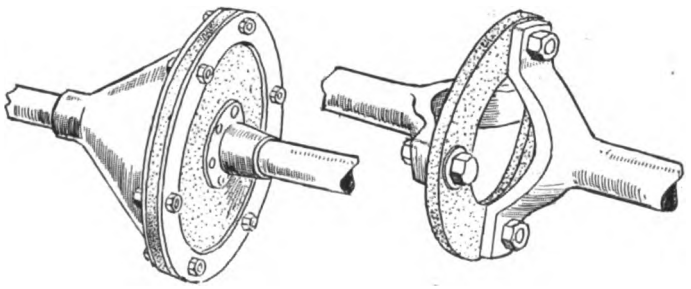

FIG. 12—TWO DESIGNS OF LEATHER COUPLINGS

Left—Original That Stressed the Leather Highly, and Right—Modern Design in Which the Leather is Only in Direct Tension

for driving an overhead camshaft, where its facility for being taken apart is advantageous. It is a joint permitting neat enclosure but it is rather difficult to keep down the bearing pressures without using large and heavy parts in its construction.

In Fig. 7 is shown a joint used by the Wolseley Company for some seven or eight years. The outer ring is split transversely and each of the pins is bushed. Thin grease or heavy oil can be supplied through a screw in the outer case. It will be noticed that the universal ring is integral with the cap. A more usual construction for what is commonly called a ring joint is shown in Fig. 8, this being typical of a great many Continental designs. Here the ring is split in a horizontal plane. It is not usual to find this type made integral with any part of the enclosure.

Fig. 9 shows the enclosure scheme adopted for a well-known British car, an example of the stationary type of cover. This differs somewhat from the stationary case described previously, in that the brass cone seen on the propeller-shaft is fixed to the latter and there is no connection between it and the stationary brass case enclosing the joint. The purpose of the part on the propeller-shaft is to protect the opening to the main case. This eliminates the necessity for a

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slip-ring on the propeller-shaft and has worked out very well in practice, the cone protecting the opening into the main case sufficiently to exclude practically all dirt and water.

Fig. 10 shows a rather ingenious way of enclosing a DeDion type of telescopic joint. This idea has been developed in many different ways, the illustration being taken from a drawing of a British light car.

COMPROMISE REQUIRED IN JOINTS

Consideration of universal joints, as well as of their enclosure, indicates the need for compromise so often found in automobile construction. From the standpoint of the universal joint there is no

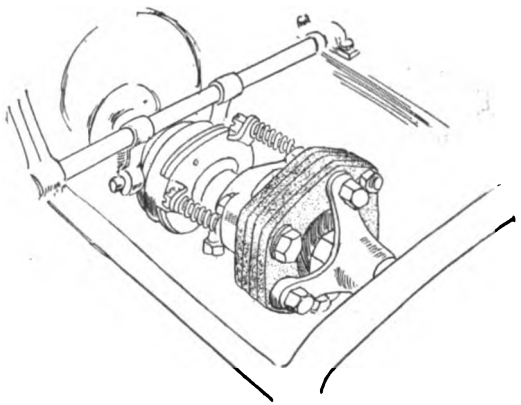

FIG. 13—LEATHER JOINT USED BETWEEN CLUTCH AND GEARBOX

better enclosure than that provided by a torque-tube with a single joint at the front end of the propeller-shaft, contained within a spherical housing formed on the end of the torque-tube and set into a corresponding socket on the back of the gearbox. A construction of this sort, Fig. 11, allows copious lubrication from the gearset and, in practice, appears to insure an almost everlasting universal if all the details are carried out well.

Against this design, however, we have the velocity variations caused by the action of a single joint, and also the weight and cost of the torque-tube construction. At present it appears that the simplicity and lightness of the Hotchkiss drive, or at least of a double-jointed shaft with a light torque stay, more than outweigh the better protection of the torque-tube. There is no easy way of lubricating universals on an open-type shaft except by periodical grease packing. Thus automobile designers must choose joints of such size that they will not be overloaded and will contain a good supply of lubricant. Picking the smallest size of joint that the manufacturer will guarantee

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for the car is poor policy, as it means poor durability in any case and very poor service indeed if the owner of the car neglects to grease up frequently; and he usually neglects the universals more than any other point on the chassis.

FUTURE OF FLEXIBLE MATERIAL

A type of joint that has tempted inventors for ten years and more is that in which there is no bearing or working surface whatever. Instead of a joint a sheet or ring of flexible material is employed, the natural elasticity of the substance being used to give the necessary movement.

The most common coupling of this sort is the leather disk so often used for driving magnetos or other engine accessories. Leather ring-

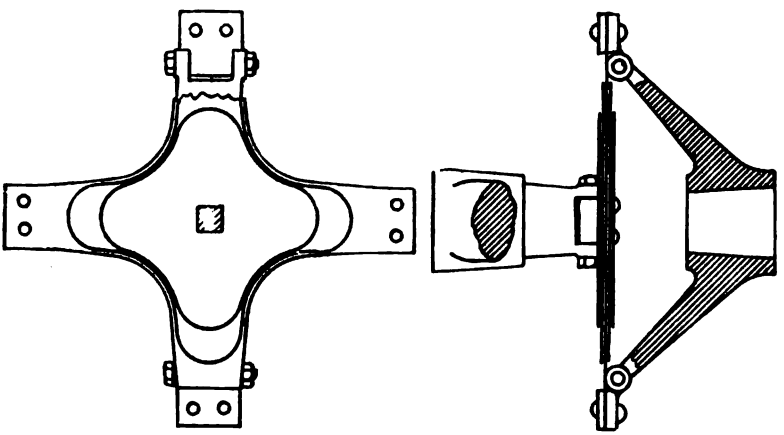

FIG. 14—DESIGN FOR UNIVERSAL JOINT MADE ENTIRELY OF SPRING STEEL

couplings like Fig. 12 are commonly used between the clutch and the gearset in European designs, and on some American cars, notably the Jeffery, Fig. 13, and they have been tried for the propeller-shaft. At present no American car has propeller-shaft joints of this character, but several were on the European market just before the war.

The heaviest service to which leather ring-joints have ever been put is on the omnibuses in London. Owing to variations in the load the universals on an omnibus see heavy service and the noise of a worn pin-joint is objectionable. At first the leather joints seemed to be ideal. They lasted for a good many thousand miles, were cheap, more easy to replace, and always extremely quiet. However, the latest advices from London are that the leather is being abandoned in favor of joints with ball-bearing or roller-bearing pins, as these appear to last forever without wear, maintaining the quietness of leather

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without the need for periodical renewal. This is in heavy service. It seems possible that we have not yet heard the last of flexible ring-joints for passenger-car service. Leather rings will take all the drive a passenger-car engine can supply, and their drawback is mainly liability to deterioration by the action of grease and water, which slowly rot the material. To overcome this difficulty, disks are now being made of oilproof and waterproof substances like woven asbestos, and the trials are encouraging, though it is early yet for prediction.

Another style never developed to the full in actual practice is the joint, Fig. 14, using spring steel instead of leather or fabric. Such joints usually consist of a series of thin spring-steel rings clipped together and mounted exactly like a leather ring. In this form they have proved excellent for clutch couplings, but are probably no better than leather, while considerably more costly. Also a good large leather joint will permit sufficient end-displacement to allow for clutch withdrawal without the provision of a metallic sliding joint; the spring-steel ring will hardly do this.

Several patents have been taken out for special designs using thin steel leaves to give freedom both axially and angularly. Some of them appear promising on paper but only hard service can decide their merits.

CONCLUSION

This paper is not intended to suggest that the universals we have to-day are other than excellent for their purpose. On the contrary, considering how bad universal joints were a short time ago and how little trouble they give in a modern car, and, still more, considering the price at which they are sold, the modern joints are usually wonderfully good. As in the case of all other parts of the automobile, however, there is no reason to suppose that we shall *never* have anything better, and this brief examination of the subject will have served its purpose if it draws attention to some of the problems connected with the provision of an efficient, silent and durable universal construction that will maintain itself without attention from the driver of the vehicle. It appears to the author that the American automobile engineer owes a great debt of gratitude to the specialists who have taken upon themselves the task of caring for a part of the chassis that has been extremely troublesome to foreign engineers.

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DETROIT SECTION PAPERS

EVOLUTION OF REAR AXLE

By J. G. Perrin

(Member of the Society)

Abstract

The author limits his consideration of rear axles to that of the bevel-gear type and takes up the subject under the heads of load-carrying member, gearing, driving-shaft, brakes and materials. Several different forms of cast and pressed axle housings are briefly described. Mention is then made of the axle gearing and data given showing the end thrust for straight-tooth bevel pinions. The methods of supporting pinions and of attaching bevel gears to the differential are discussed.

Forms of differentials are considered, several different conventional types being illustrated. The subject of driving shafts is briefly reviewed, as is also that of brakes and brake materials. The author concludes his paper by figures showing the tensile strength, elastic limit and elongation of the metals used in the various parts of the rear axle and also explains some oil-retaining and dust-protecting features of design.

The axle and wheel are components of the modern motor car that have always had some equivalent in vehicles since the rolling type succeeded the sled as a medium for transportation. Since the axle has always been in existence the present form is a matter of development and evolution rather than invention.

In some of the earlier forms of motor-propelled vehicles a belt or chain was used for transmitting the power from its source to the driving wheel. Owing to the number of parts, the difficulty of arranging a satisfactory form of housing for the chain, and the loss of power from belt and other forms of friction drives, these systems have been succeeded by some type of positive gear drive. Figs. 1 and 2 show early forms of single- and double-chain drive.

On pleasure-car work the bevel-gear axle has up to this time been an example of the survival of the fittest, although the worm is favorably considered when low gear reductions are required. For the purpose of convenience I will describe the development and the present form of the bevel-gear axle under the heads: Load-carrying Member, Gearing, Driving-shafts, Brakes, and Materials.

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LOAD-CARRYING MEMBER

The load-sustaining portion of the rear axle has passed through a greater number of stages of development than any other part of the unit. Fig. 3 shows an early form in which the axle housing is divided through a horizontal center line. The two halves are bolted together and at the same time the tubes extending to the right and left are clamped to the spring seats by upset flanges, or collars, on the end of each tube.

Fig. 4 shows two straight tubes driven tightly and riveted into a malleable-iron center casting that has an opening in the upper half for assembling the differential and gears. This opening is sealed by a cover fitting over the ends of the axle housing and clamped to the main body of the casting along a horizontal center line.

The construction shown in Fig. 5 is fundamentally the same as in Fig. 4, with the exception that the parting line for the cover is above the cylindrical ends of the center casting so that the cover is seated along a straight line and can be fitted more easily to keep oil tight.

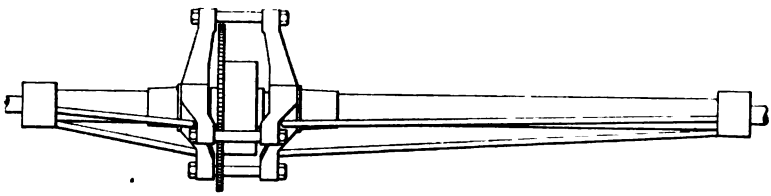

FIG. 1—AXLE ILLUSTRATING SINGLE-CHAIN DRIVE USED IN EARLY AUTOMOBILES

In the form shown in Fig. 6 the side tube members open in a funnel shape that fits over and is riveted to a corresponding surface on the center casting. The cover is fitted on a flat seat in a position far enough above the center line of the axle to permit the full engagement of the funnel-shaped tube ends with the center casting. This is a neat construction but puts a great deal of strain on the rivets.

In Fig. 7 the center casting is divided through a vertical center line and tubes are driven in and riveted to the sides of the center housing.

Pressed-Steel Housings

Fig. 8 shows one of the early forms of pressed steel housings, consisting of two central stampings welded through a vertical center line. The tubes on the end were welded again to the center, making three welded joints in vertical lines. The stamping is closed at the rear and the differential and driving gears are self-contained in a malleable-iron casting, or carrier, which is bolted along a vertical line to a turned-over flange on the steel stamping. The inaccessibility

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of this construction resulted in a steel stamping, Fig. 9, symmetrical as regards both horizontal and vertical center lines. The carrier casting is bolted to the front of this stamped housing and a cover of either cast or pressed steel bolted to the rear. This general design is probably used by more car makers than any other form of axle construction. The adaptability of this housing is shown by the fact that it is used on cars ranging in price from $395 to the highest in this country, the only difference being in the size of stamping and thickness of steel. Most of these housings are stamped from a good

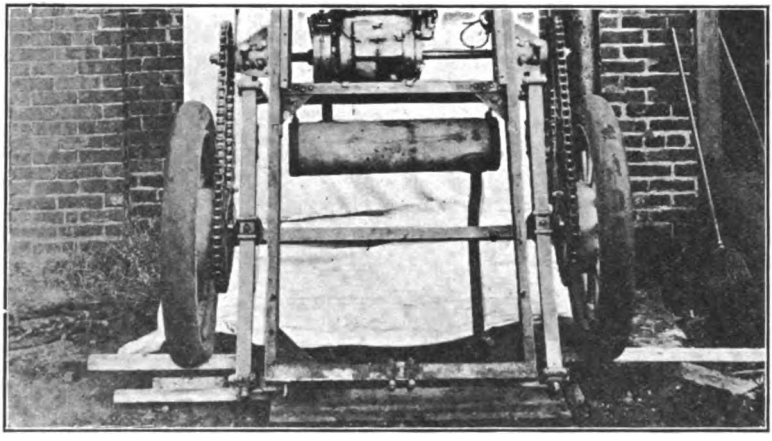

FIG. 2—CHASSIS EQUIPPED WITH EARLY FORM OF DOUBLE-CHAIN DRIVE

grade of carbon steel of the right composition for press work. They are made as a rule of two stampings symmetrical as considered from a horizontal or vertical center line.

The housing in Fig. 10 has the division line horizontal. The two halves are welded together and the ends turned for the assembling of brake spiders and spring seats.

Another form of vertical seam is shown in cross-section, Fig. 11. The edges of the metal are opened into a slight V to give a good chance for the welding material to flow. This form of joint is necessary for welding where the material thickness is 5/32 in. or less, as the material is then hard to weld in a straight butt joint.

Fig. 12 shows a recent development of the pressed-steel housing in which a square tubular section is used under the spring seat and the ends flared up for brake spider anchorage. This form gives a secure spring seat anchorage for Hotchkiss drive.

On the larger housings for cars weighing about 4000 lb. the material used is 3/16 in. thick, and the butt joint can then be success-

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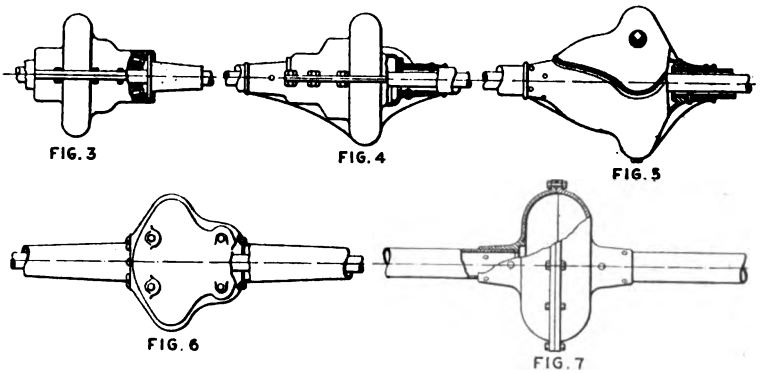
fully welded on a horizontal line. These housings range in weight from 30½ lb. for one made of ⅛-in. metal for a light car in which the springs are supported at the extreme ends of the housing, to 55½ lb. for a design of ¼-in. material for 5000-lb. cars and 54½ lb. for one 3/16-in. thick for 4000-lb. cars. The last weight includes sleeves. The 55½-lb. housing is made of tubing, so that the final form has no weld whatever. This construction proved so expensive that it had to be given up for a more commercial method.

The proportions of the pressed-steel housing are determined by the weight of the car and the size of the driving bevel gears. The arch in the center has to be made with an opening large enough to assemble the bevel gear, the size of which is determined by the power to be transmitted and the gear reduction required. The load-carrying capacity of the housing is determined by the diameter of the ends of



FIGS. 3-7—FORMS SHOWING DEVELOPMENT OF CAST AXLE-HOUSINGS

the housing. These diameters generally range from 2½ by 3/16 in. for 2000-lb. cars to 3 by 3/16 in. for 4000-lb. and 3½ by ¼ in. for 5000-lb. cars.

The ability of these housings to resist torsional stress is shown by actual tests in which the bare housing showed a torsional moment of 27,000 lb. in. at the elastic limit of the material and of 50,000 lb. in. at the breaking load. These were made on a medium size housing without the carrier or cover bolted to the center, which greatly strengthen the central portion of the housing.

GEARING OF THE AXLE

Until about two years ago the straight-tooth was the prevailing form of bevel drive gears. With improvements in engines and transmissions such that these units operated more quietly the difficulties became greater in getting straight-tooth bevel gears as noiseless as

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the trade demanded. This condition was aggravated by the increase in production which made it important that the gears operate as noiselessly as possible with a minimum of adjustments, fitting and trying after a chassis was put on the road. Then the gear reductions, on account of greater range of speed on direct drive, necessitated either small pinions or larger driving gears. These again added to

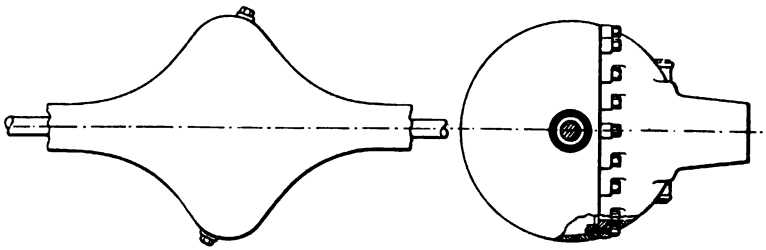

FIG. 8—PRESSED-STEEL AXLE-HOUSING MADE OF WELDED PARTS

the difficulties of getting quiet gears. A solution of this difficulty was sought in two directions—the use of the spiral bevel and of the worm gear.

The worm gear drive, Fig. 13, solved the difficulty, but the problem of its production in sufficient quantities in a commercial way was not solved as quickly as was that of the spiral bevel gear. The spiral, or helical, gear permitted use of standard parts with but little tool

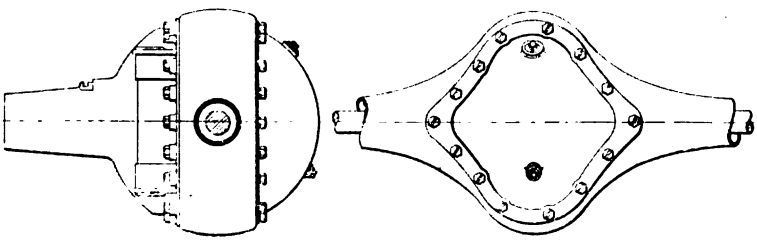

FIG. 9—WIDELY USED FORM OF PRESSED-STEEL AXLE-HOUSING

expense, which was also a strong reason for its adoption. This form of bevel tooth came into use just as fast as machinery could be made to produce it and is now a commercial possibility for not only the high- but even the lower-priced cars.

The use of small pinions to get a large gear reduction has resulted in the pinion being made integral with the pinion shaft in the majority of installations. The spiral bevel operates satisfactorily with eleven- and even ten-tooth pinions, and in only a few cases in the larger cars has it been necessary to make the main driving bevel so large that more than eleven teeth are needed in the pinion. The use of the spiral

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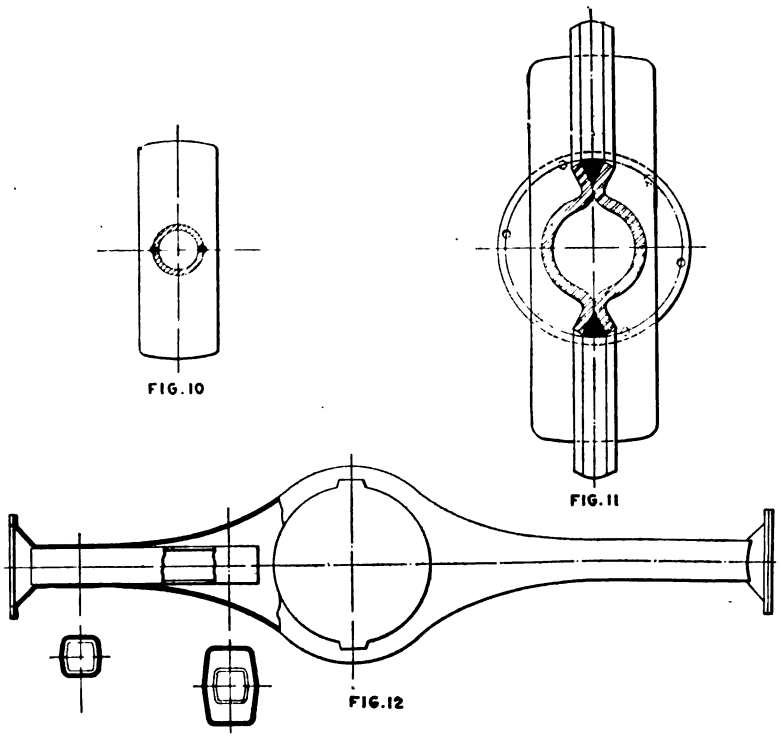
bevels has resulted in several detail changes in the mounting of the pinion-shafts and differentials, as the tendency for the pinion to screw in or out of the main-drive bevel gear necessitates greater care in providing for end-thrust in both directions, especially on the pinion-shaft bearings. Bearings capable of satisfactorily taking end-thrust as well as radial load have solved this proposition most satisfactorily.

End-Thrust of Pinions

In a typical case the end-thrust from a straight-tooth bevel pinion, ratio of 15:53, averaged 7.61 lb. for a 100-lb. tooth load, whereas a



FIGS. 10-12—THREE TYPES OF STAMPED AXLE-HOUSINGS

spiral bevel of the same ratio showed about a 70-lb. end-thrust for a 100-lb. tooth load. This was with about a 30-deg. angle of spiral.

A few years ago gear ratios of 3:1 and 4:1 were usual practice. To-day ratios are more commonly 4½:1, and demands for reductions of 5:1 and even 5½:1 are becoming more and more pressing.

While ten-tooth pinions are in use, they are to be avoided on

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account of the more unfavorable tooth form and the greater tooth pressure, unless the pitch can be increased to take care of greater strain. Even the eleven-tooth pinion would not be possible without the use of the spiral-cut tooth, of better tooth profiles, as well as of better materials.

The use of a pinion-shaft extended forward, surrounded by a tube that is fastened rigidly to the axle housing and is supported at the front end so as to take the torque and sometimes the drive also, seems to be gradually giving place to the short pinion-shaft with two universal joints on the propeller-shaft. In this latter arrangement the torque is taken on arms fastened to the axle housing rigidly or

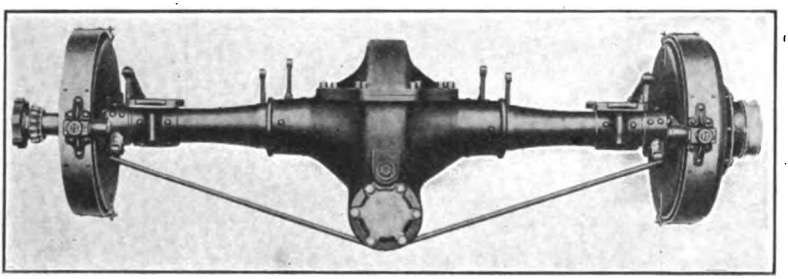

FIG. 13—COMPLETE ASSEMBLY OF MODERN TYPE OF WORM-GEAR AXLE

arranged to swing in a horizontal plane; or else through the medium of side springs that have their seats fastened rigidly to the axle. The construction using the springs is commonly called the Hotchkiss drive (from the name of the Hotchkiss Company of France, which first introduced it) and is increasing in use.

Methods of Supporting Pinions

The arrangement of bearings to support the pinion-shaft has three general forms; one in which the pinion is supported between bearings; another in which the pinion has a bearing directly back of it that takes both radial and thrust load and with the other bearing at the extreme end of the concentric tube or third member. The second form gives a wide separation of bearings, but is not used very much. The third and most common form is that having the bearing a short distance forward, the distance being determined by the proper distribution of load on the two bearings. In fact, the length of pinion-shaft is determined by the proper spacing of bearings to get proper distribution of load.

As stated, the use of a small number of teeth in the pinion has made almost necessary the integral pinion-shaft and pinion. Where the pinion is made separate it is generally secured to the shaft by a

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taper-fit, key and nut. The universal-joint flange, or coupling, is generally fastened to the front end of the pinion-shaft in the same way, both tapers being S. A. E. standard. Other methods of securing the coupling flange to the pinion-shaft are by means of square and spline fittings.

Main-Drive Bevel Gear

The large bevel gear used to be a considerable problem as regards forging, proper and uniform hardening and straightening after hardening. With improved methods—proper selection of steel, uniformity of steel, holding the gear in shape during the quenching

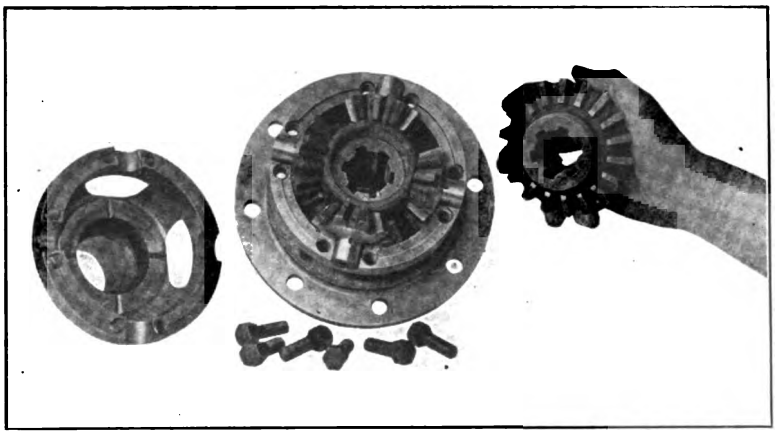

FIG. 14—SHOWING PARTS OF CONVENTIONAL BEVEL-GEAR DIFFERENTIAL

operation, ingenious straightening methods and accurate fixtures for grinding the seat of gear true in relation with the pitch line—this problem has been satisfactorily solved. This gear is fastened to the differential flange either by riveting or bolting. A satisfactory way has been found for riveting the gear in place, and this method is now used in large production. The riveting is done cold, either by hand or by power press, the latter method having been developed to a point where it is reliable and results in seating the driving bevel uniformly against the flange, giving a true-running condition.

Forms of Differentials

The differential itself has remained fundamentally the same for several years, being constituted of a nest of either spur or bevel gears, the bevel type predominating. The usual form of bevel-gear differential consists of a spider having two, three, or four trunnions,

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these being clamped at their ends between the two halves of the differential casing. The pinions revolve on these trunnions, giving a good bearing condition of hardened steel surfaces. Sometimes a bronze bushing is inserted in the pinion, but in the majority of cases no extra bushings are needed, since the hardened pinion runs well on the hardened spider pins. A conventional form of bevel gear differential is shown in Fig. 14.

A differential developed to meet cost reduction has a one-piece casing with two pinions, Fig. 15. The case is a one-piece malleable casting. The gears and spider are assembled and the trunnions on

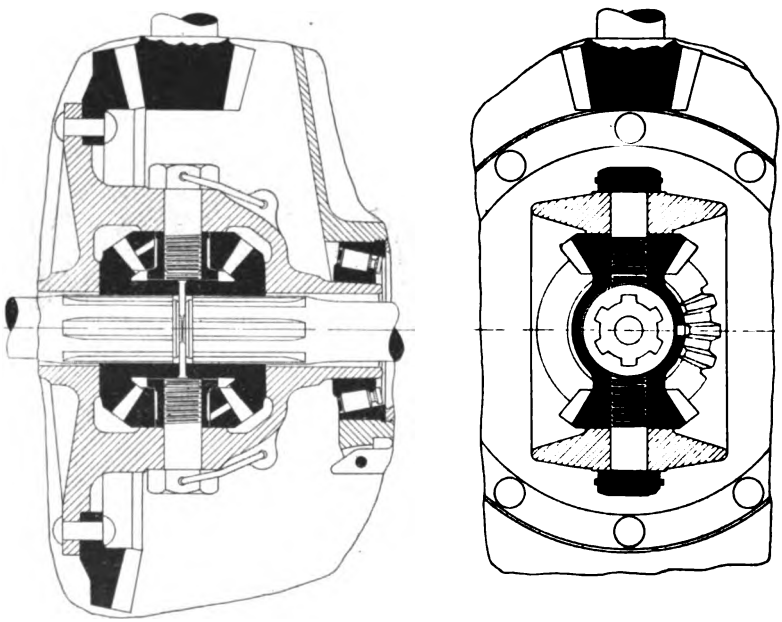

FIG. 15—DIFFERENTIAL WITH ONE-PIECE CASING AND TWO PINIONS

the spider anchored by screwing hardened bolts through the outside of the casing into the spider. Some little juggling is necessary in laying out one of these differentials to get compactness and ease of assembling. It must be possible to assemble with the main bevel gear riveted in place on the differential-case flange, as the teeth of this gear usually overhang the differential side-gear. This design enables each wheel bearing in a semi-floating axle to take inward side-thrust from the opposite wheel by passing the thrust through the driving-shafts, which butt with hardened buttons as points of contact.

Recently the positive-drive form of differential in which the drive

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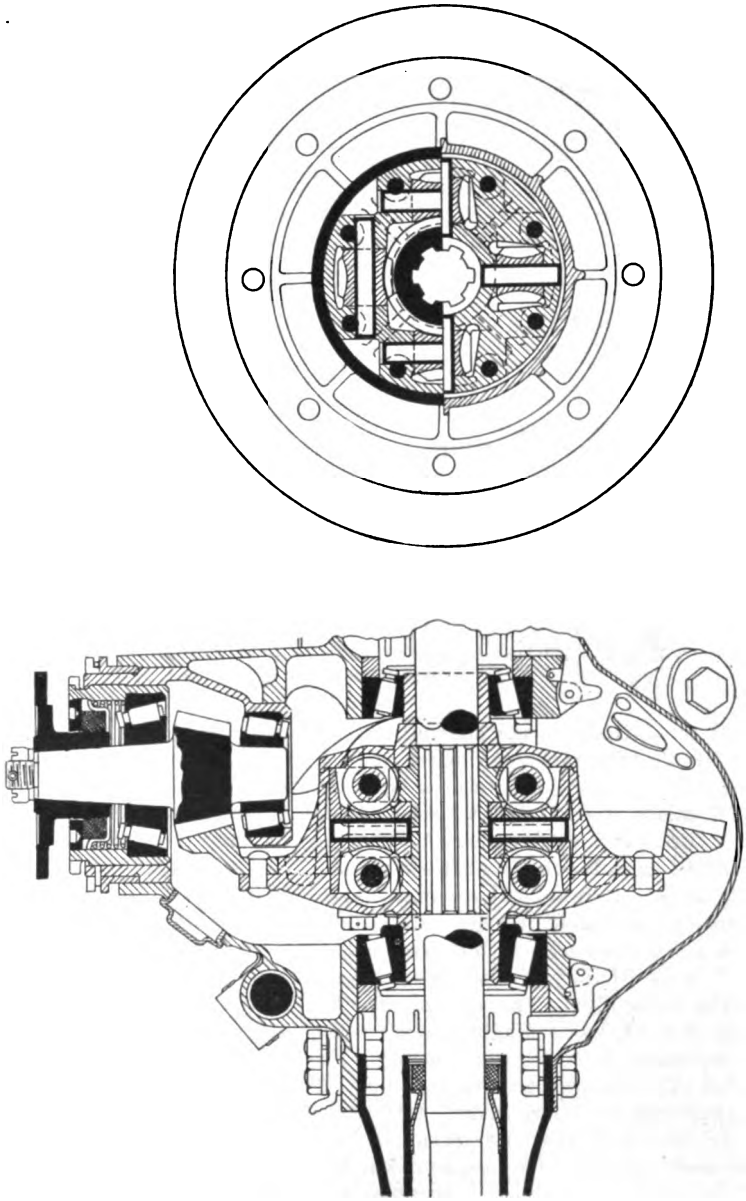

FIG. 16—DIFFERENTIAL GIVING POSITIVE-DRIVE EFFECT ON WHEEL GIVING BEST TRACTION

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is taken on the wheel giving the best traction has been developed commercially in several forms. One of these is illustrated in Fig. 16. The advantage of this construction is due to the increased friction of the differential pinions. The elimination of the differential is a possibility suggested from experiments a number of people are now conducting on both commercial and pleasure vehicles.

DRIVING-SHAFTS

Driving-shafts can be classified as either full-floating or fixed-hub. In the full-floating type, Fig. 17, all the bending stress due to static load and skidding force is carried by the housing, while the driving-

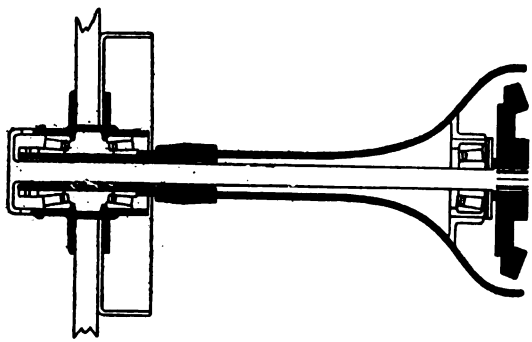

FIG. 17—FULL-FLOATING AXLE IN WHICH SHAFT TAKES DRIVING EFFORT ONLY

shaft turns freely in the housing and bears only the stress of turning the wheel. The inner end of the shaft is driven from the differential side-gear by either splines or a square fitting. The outer end of the shaft drives the wheel hub either through a flange formed integrally with the shaft, the periphery of the flange having teeth that engage with corresponding teeth on the wheel hub, or by a flange secured to the axle shaft by keys, splines or a square fitting. The wheel hub is supported independently on the housing by a pair of bearings spaced such a distance apart as has been found best practice.

The fixed-hub type takes two general forms, both having the wheels attached rigidly to the driving-shafts. The so-called semi-floating type, Fig. 18, has a single bearing fitting over each shaft and inside an extension from the axle housing. Fig. 19 shows what is commonly called the three-quarter floating type. The inner race of the bearing is supported on an extension of the housing and the hub so designed as to revolve on the outer race of this bearing. In both these types the shaft transmits the torque and also supports the weight of the car. All three types of axle are in common use, and giving satisfaction, with the same care used in the design and selection of material.

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REAR AXLE BRAKES

These are of two general types, contracting and expanding. (Several disk brakes have been designed, but none is in general use.) Pneumatic, hydraulic and other forms of brakes have been used, but none of them has been developed commercially. Contracting brakes are generally used on the same drums with internal expanding brakes. Fig. 20 shows such a combination.

The contracting brake-band in common use consists of a band of steel to which is riveted a lining in which asbestos fiber has been woven. This band material is 0.109 in. or more thick and varies in width on pleasure cars from 2 to 2½ in. Practical examples of contracting bands in satisfactory service show that the braking surface for each 100-lb. weight of car is 5.9 to 6.5 sq. in. on a 4000-lb. car and 5.7 sq. in. on a 3000-lb. car. The lining material is riveted to the steel band

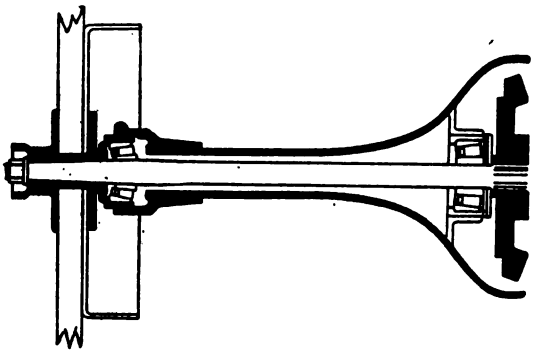

FIG. 18—SEMI-FLOATING AXLE WITH BEARINGS INSIDE HOUSING

by copper rivets in the best practice and by steel rivets in the lower-cost production.

The brake-band, as a whole, is generally anchored at one side and the contracting levers at the opposite side. Contracting brakes are also made with forged or cast bands or shoes hinged at one or two points where they are anchored. The lever action is almost invariably of the double-acting type.

There are more varieties of methods used in bringing the band in contact with the brake-drum in internal brakes than in the contracting or external type. Probably the simplest and most common form of internal brake, Fig. 21, is operated by a cam. The brake-band, or shoe, is anchored at one point and the ends of the band on the opposite side are fitted with brackets between which the operating cam turns, opening or expanding the brake-band against the drum. A strong retracting spring is used to hold both the ends of the shoes in contact with the cam and the band out of contact with the drum. The shoes

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can be made either of a piece of heavy band steel, which is generally continuous, or of forgings or castings, which are hinged at the point of anchorage to the brake spider.

In nearly all cases the same lining material is used for shoes for internal brakes as for the contracting brake for pleasure vehicles. In the use of cam brakes provision should be made for getting a proper length of lever on the end of the shaft that operates the expanding cam. With a properly designed cam and an adequate length of lever the cam brake operates as successfully as any other form of internal brake, considering simplicity and efficiency.

An internal brake operated by toggle joints, Fig. 20, is effective, but has more parts and costs more to produce than the cam brake.

Generally speaking, foreign practice has always favored the transmission or propeller-shaft brake for service work and a rear axle brake for emergency work. American car builders, however, have

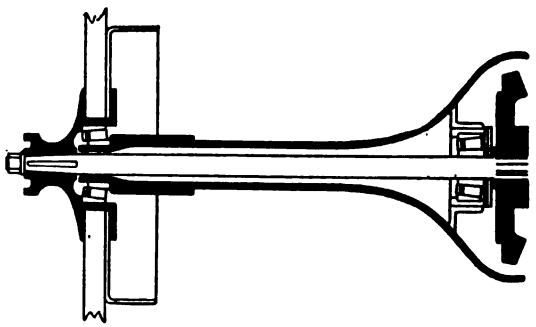

FIG. 19—THREE-QUARTER FLOATING AXLE WITH BEARINGS BETWEEN HOUSING AND HUBS

almost always installed both service and emergency brakes on the rear axle, although there are cases of American car makers using one brake on the rear axle and one between the transmission and rear axle. The placing of both brakes on the rear wheels is a great advantage from the car assembler's viewpoint. In order to get a powerful brake, easily operated by women drivers, a few manufacturers have used a propeller-shaft design. In using this brake, the car makers should consider the additional strains on the propeller-shaft joints, bearings and driving gears. Both systems of braking can be made entirely satisfactory, dependent upon the general design of the brakes and the method used to transmit the effort of the drive in applying the pedal or lever.

One interesting method, Fig. 22, of getting a geared-up brake without having the braking stress pass through the universal joint is used by the La Buire Company of France. In this design the brake

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drum is mounted on the pinion-shaft. Fig. 22 shows the outside appearance of this axle, and Fig. 27 the method by which the pinion-shaft is passed through the axle housing. The latter does not show the brake drum in place, but it is simply a matter of passing the pinion-shaft through the rear end bearing and of taking care of the installation on the brake drum.

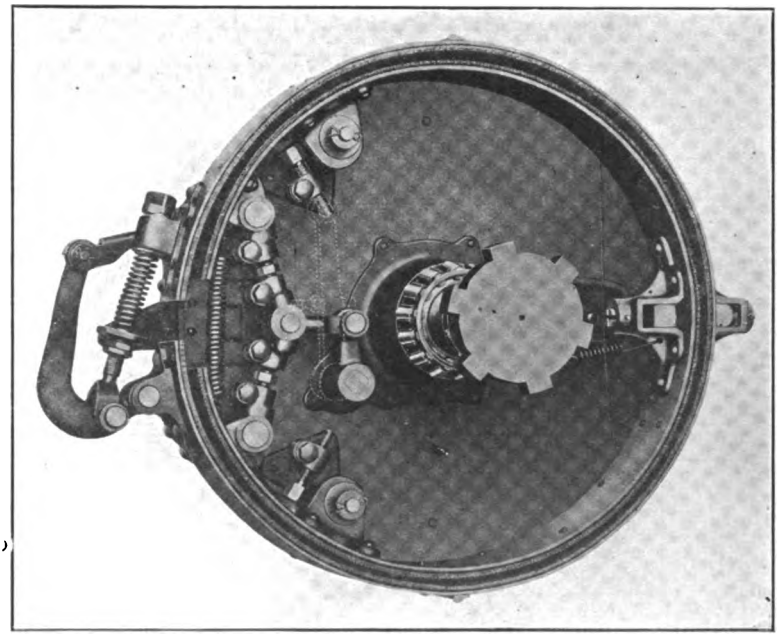

FIG. 20—CONTRACTING AND EXPANDING BRAKES MOUNTED ON SAME DRUM

MATERIALS USED IN REAR AXLES

Malleable iron is used in differential housings in the carrier units that hold the pinion-shaft and differential bearings, brake spiders and some of the hubs. This material has the following average physical properties:

Tensile strength	43,790 lb. per sq. in.
Elastic limit	30,540 " " " "

Drive-shafts are made from chrome nickel steel with these physical properties:

Tensile strength	145,000 lb. per sq. in.
Elastic limit	120,000 " " " "
Elongation in 2 in.....	15 per cent
Reduction of area.....	52 " "

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This material in a shaft of 1 5/16-in. diameter, the size used in large production, when subjected to torsion shows the following ability:

Torsional moment	
Tensile strength	54,000 lb. in.
Elastic limit	33,000 " "
Angle of twist.....	1,300 deg.
Fiber stress	74,600 lb. per sq. in.

The sleeve used in the full-floating type of axle is made of either

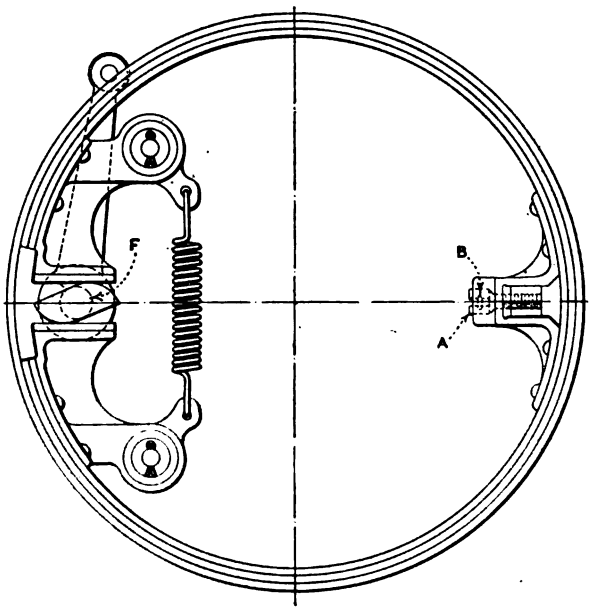

FIG. 21—CAM-OPERATED FORM OF INTERNAL BRAKE

carbon or chrome nickel steel, according to the size and load capacity of axle. The chrome nickel steel shows the following physical properties:

Tensile strength	105,500 lb. per sq. in.
Elastic limit	81,000 " " " "
Elongation in 2 in.....	25 per cent
Reduction of area.....	66 " "

The bevel gear and pinion have to be of a steel of high physical

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properties with a hard wearing surface and tough inner core. This necessitates the use of a case-hardening low-carbon steel alloyed with chromium and nickel. The core shows the following average physical properties:

Tensile strength	107,000 lb. per sq. in.
Elastic limit	60,000 " " " "
Elongation in 2 in	22 per cent
Reduction of area	40 " "

This steel, when heat treated for use as gears, with a hard surface and soft core, shows the following physical characteristics:

Tensile strength	116,000 lb. per sq. in.
Elastic limit	107,500 " " " "
Elongation in 2 in	1.0 per cent
Reduction of area	1.5 " "

The physical tests of case-hardened specimens vary a great deal,

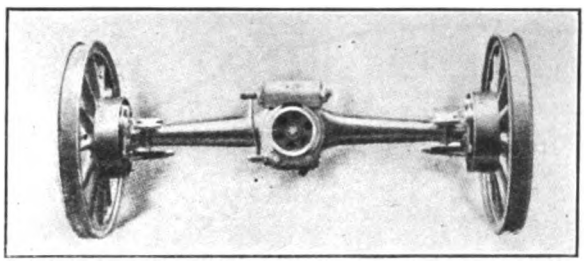

FIG. 22—LA BUIRE GEARED-UP BRAKE WITH DRUM ON PINION-SHAFT

so that we have to judge them by the results actually obtained rather than by the tests.

OIL-RETAINING AND DUST-PROTECTING FEATURES

One of the problems has been to keep the oil in the center housing used for lubricating the gears, differentials and bearings from working out onto the brakes. This has been successfully overcome in the case of the full-floating type of axle by prolonging the sleeve, see Fig. 23, to a point as near the differential bearings as possible. In the fixed-hub type of axle the scheme shown in Fig. 24 takes care of this satisfactorily by having a felt washer at the end of the driving shaft, just inside the bearing.

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EFFECT OF ENGINE DEVELOPMENT

The subject is sometimes brought up regarding the effect on axle design of the use of high-speed six-, eight- and twelve-cylinder engines of small bore. When practically the same chassis as regards passenger-carrying capacity is used, the substitution of these engines for

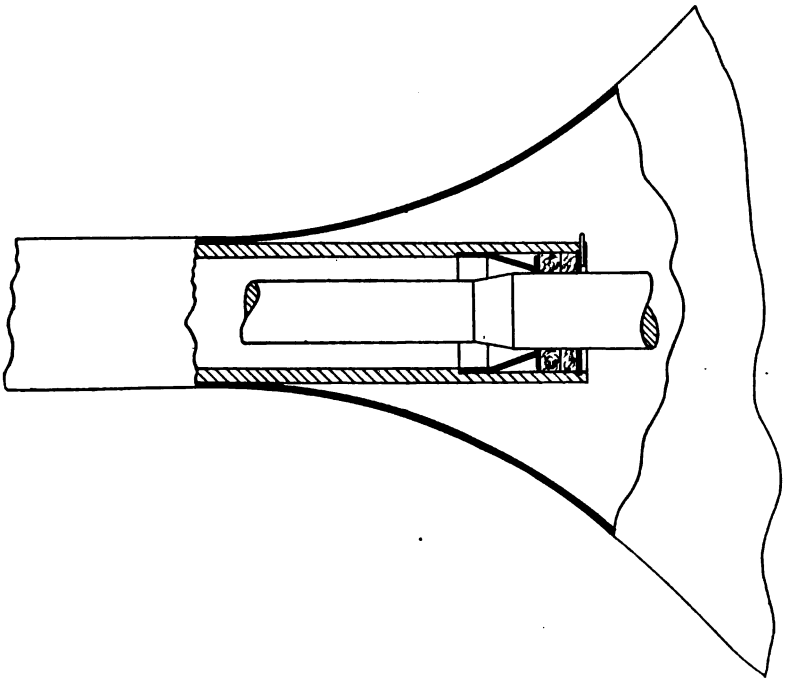

FIG. 23—SHOWING SLEEVE PROLONGED TO KEEP OIL FROM BRAKES

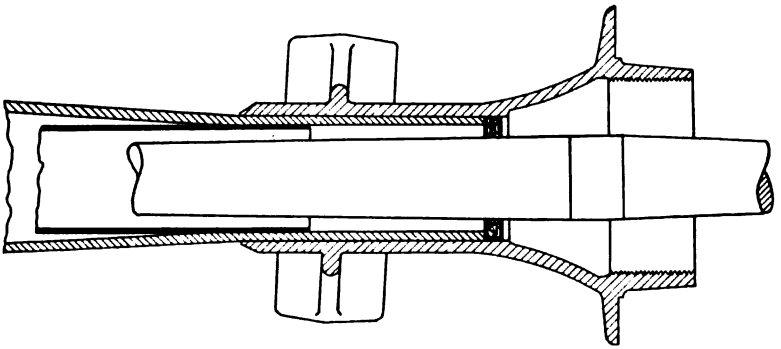

FIG. 24—OIL RETAINING FEATURE IN FIXED-HUB TYPE OF AXLE

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the larger-bore engines of a smaller number of cylinders has resulted in no change in the fundamental dimensions of axle as regards size of drive-shaft, bearings and driving bevels.

The torque of the high-speed engine is generally better than that

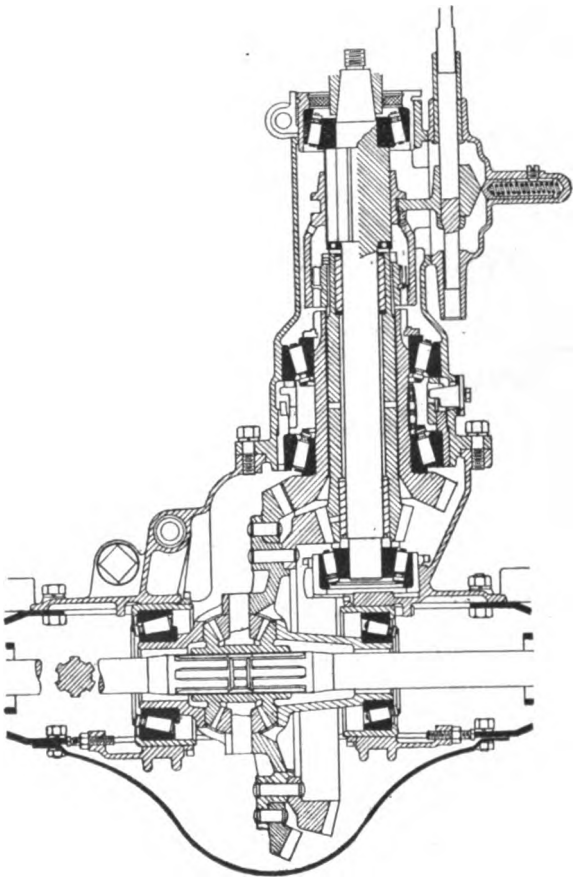

FIG. 25—TWO-SPEED TYPE OF REAR-AXLE REDUCTION

of some of the former slower-speed engines. The problem, therefore, resolves itself into a question of torque. An engine with a torque of 2000 lb. in. from twelve cylinders is capable of exerting as much strain on the axle-driving parts as a four-cylinder large-bore engine of similar ability.

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UNUSUAL TYPES OF AXLES

Before closing, it would be interesting to consider a few unusual types of axles. There are innumerable rear-axle constructions, yet the particular types illustrated are distinct and have been used by leading makers in Europe and this country.

The two-speed axle, Fig. 25, has been used in large quantities. It enables a direct drive of large gear reduction to be used for traffic

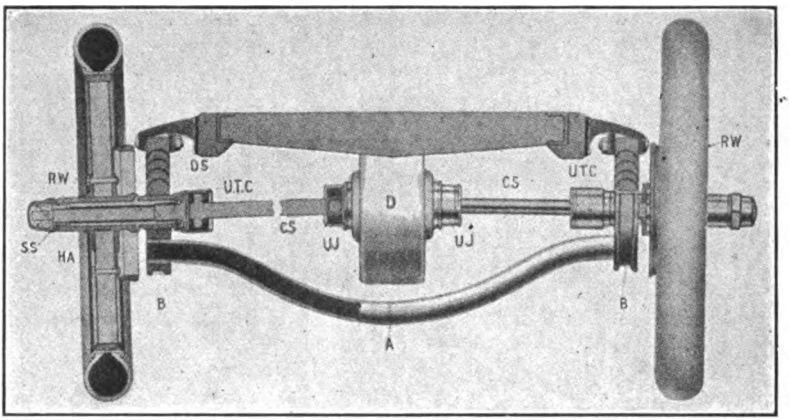

FIG. 26—EARLY TYPE OF DE DION REAR-AXLE IN PART SECTION

A, tubular axle; B, axle bracket; H A, hollow axle; R W, road wheel; S S, square socket; U T C, universal telescope coupling; C S, driving-shaft; U J, universal joint; D, differential gear-box.

work and a lower reduction for country where stretches of good road permit high car and reasonable engine speeds, resulting in better fuel economy.

Fig. 26 shows an early attempt by the De Dion Company to reduce unsprung weight and to lessen both the effects of road shocks on bearings and the number of driving-gear adjustments required. The same company later brought out an axle of conventional type, indicating the design shown was unnecessary.

Further details of the La Buire axle, Fig. 27, sho wthe method used in carrying the pinion-shaft directly through the center of the axle housing. The advantage claimed for this construction is the possibility of cambering the rear wheels, of the use of a lighter differential, and of installing a propeller-shaft brake that does not strain the universal joints.

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DISCUSSION

CHAIRMAN GEORGE W. DUNHAM:—The rear axle is probably the most abused part of the motor car. It must carry the car load; occasionally the load comes down on the bumper, subjecting the axle to enormous strains. It is also subject to unsprung road shocks. The rear axle transmits the torque from the driving power, and also has to do the work of propelling the vehicle, which is a strain in still another direction. Then it is subject to side strains from the wheels, as the car skids, drops into ruts or strikes sudden obstructions. Furthermore, it is doing this work in the dirt and mud at all times. There

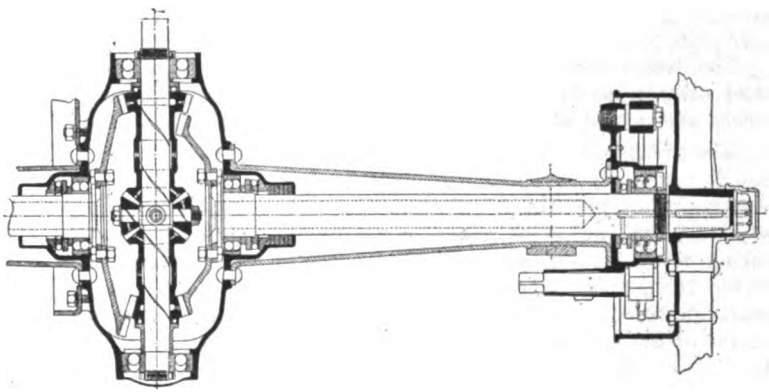

FIG. 27—LA BUIRE AXLE WITH PINION-SHAFT EXTENDING THROUGH HOUSING

is still much opportunity of development in an axle. Only recently have oil leaks been handled so that they are under control. Many axles built to-day are still subject to oil-leak troubles.

POSSIBILITIES OF ELIMINATING DIFFERENTIALS

CECIL H. TAYLOR:—Is it feasible to run axles without a differential?

J. G. PERRIN:—The elimination of differentials is a matter that is being experimented with all over the country. Abroad a number of army ambulances are being operated without differentials. It is found that the efficiency of trucks is increased a great deal in straight-ahead travel. They cannot turn around in close quarters on soft, rough ground on account of the cramping effect on the front wheels. The only objection raised so far in this country has been the fact that in turning around in a short space the speed is slightly retarded and the strain is, of course, all put on one shaft.

One company building electric vehicles has tried out in New York

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City driving a car with the differential locked. The ampere draw was practically the same as in ordinary running and no difference could be detected in the tire wear. On racing cars the differential has been eliminated for a long time. One of the reasons was the saving of the tires.

D. McCall White:—Is not the almost wholesale adoption of the type of axle shown in Fig. 9 due rather to the quantities being made by one company than to the fact it is the design having the maximum strength for the minimum weight? This particular banjo-casing design was originated, if I remember correctly, in England, by the Daimler Motor Car Company in 1905 or 1906 and has since been used largely in this country.

STRESSES IN AXLE HOUSINGS

The banjo-casing or pressed-steel housing is made from carbon steel with an elastic limit of 34,000 lb. per sq. in. and therefore cannot be stressed above 7000 lb. per sq. in. without disastrous effect.

The greatest stress in a full-floating axle is near the spring-seat, and is caused by the combined loads from the weight of the car, the tractive effort and the torque reaction. The diameter of the casing at this point cannot be great, so either the tubular portion of the casing must be somewhat thick to withstand the stress, or an inserted tube, made from alloy steel, must be used. This inner tube must, in turn, be carried in and adequately supported toward the center of the casing, so that the construction is comparatively heavy. In view of the large size of the part of the casing surrounding the bevel gears, the stress is extremely low. A thin casing could be employed, except for the fact that it would not be a manufacturing possibility to weld the halves of the casing together satisfactorily, if they were less than about 5/32 in. thick. In fact, were the axle scientifically designed, without manufacturing prohibitions, we could have a pressing of varying thicknesses made from alloy steel to suit the moduli of the sections employed and the stresses that come upon those sections.

ALUMINUM CENTER-PIECES OR DIFFERENTIAL CASINGS

The different forces rear axles must resist are most complex, varying from minimum to maximum with the greatest frequency. As the rear axle is unsprung weight we must exert considerable ingenuity to make this important assembly light and of maximum rigidity. Engineers seeking the greatest reduction of weight in rear axles are rather inclined towards swaged alloy-steel tubings, flanged and bolted to an aluminum center-piece or differential casing. It is doubtful whether such an axle will withstand continuous pounding on rough roads, although it be supported by a tie-rod of sufficient strength. One prominent company used an aluminum center-piece successfully,

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but reverted to malleable iron or steel castings when aluminum became so expensive. It was then confronted with the problem of the axles breaking, evidently from the pounding of the increased weight on rough roads. On the other hand, another company found that the aluminum center went to pieces, on account of fatigue, after being used about two years. This latter casing may have been stressed too highly.

My experience in Europe leads me to believe that aluminum is perfectly safe, provided it is not overstressed. Sometimes the foundry foreman feeds zinc into the aluminum, in order to overcome casting difficulties. The result is a fracture resembling Gorgonzola cheese more than anything else.

To-day we are on the threshold of an era of quieter axles, because of the introduction of spiral bevels. The need of a rear axle, the differential of which can be removed almost as easily as a spark-plug, is almost gone. Most of us mount the gasoline tank behind the axle cover, thus making an examination difficult in any case. We could design a stronger and more rigid aluminum casing if openings were no longer necessary.

In past years I have been an advocate of worm-axle design, and still believe it has a future. The worm gear-ratio should be slightly lower than the bevel in order to compensate for the slight difference in mechanical efficiency.

BEVEL-PINION TEETH AND STRESSES

I believe that no fewer than eleven teeth should be used on the bevel pinion, if one wishes a sufficiently strong and quiet construction. I am more partial to the overhung pinion than to the pinion supported on both sides. The latter seems more scientifically correct, but the former construction has always given me the quieter rear axle.

Using materials such as Mr. Perrin suggests, the stress on the pinion teeth should not exceed 22,000 lb. per sq. in., and on the axle tubes, 13,000 lb. per sq. in. These values may seem high, but the stresses due to such loads as weight of car, weight of the axle itself, reflex engine torque, reflex brake torque, tractive effort and inertia of the axle itself must also be considered.

J. G. Perrin:—Mr. White raises some interesting points, and if any car manufacturer should ever give an axle manufacturer sufficient time, I think it would be possible to work out some of those things. But when a man wants an axle to-morrow in order to build a car next week, there is no chance to design any new features. The work must be along the lines the public is demanding. An axle is used in large quantities only when it is satisfactory, both as regards cost and efficiency. The pressed-steel axle housing has proved to be the most elastic type to suit all conditions. The number of sizes

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given in the paper shows that it does not take such a great range to cover a large field of weights and horsepowers of cars.

ADJUSTMENT OF BEVEL GEARS

F. E. MOSKOVICS:—I would like Mr. Perrin to comment on the adjustable bevel gears that several companies are making.

J. G. PERRIN:—One arrangement is the screw adjustment in the meshing of the bevel gears. Spacers of variable thickness back of the pinion, and of the main bevel gear, have been tried and are used to-day in some cases. They are feasible in small production but have not proved to be a commercial possibility in large production except in special cases.

EFFICIENCY OF SPIRAL, BEVEL AND WORM GEARS

ALEX. CHURCHWARD:—What is the efficiency of spiral, bevel and worm gears on pleasure-car rear axles?

J. G. PERRIN:—Under normal conditions all show efficiencies of over 95 per cent. The spiral bevel has, of course, to have a good end-thrust bearing in order to eliminate friction from the excessive end-thrust.

GROUND CLEARANCE WITH LARGE GEARS

C. C. HINKLEY:—To me the biggest thing in axle development is the two-pinion differential, which I believe is good for any average practice. Another thing that has always bothered me is getting a large enough ring gear in a car with a small wheel, and getting sufficient ground clearance.

Mr. Perrin said the axle housings were down to ⅛ and 5/32 in. thick. All the housings I ever saw were 3/16 or 7/32 in. thick.

J. G. PERRIN:—If you employ the power the engines are developing nowadays you must have teeth of proper size. The large ratios mean large main bevel gears, as the pinion size cannot be reduced beyond a fixed minimum. If more favorable spring suspension is used and if time is given to bring out the special tools, ⅛-in. housing is possible on the smaller car.

METHODS OF GRINDING GEARS

J. N. HEALD:—The illustrations in the paper show the bevel gear with its flange—where it fits against the housing—on the back side. We, who build grinding machines and are trying to help you to get the distance accurately from the pitch line to the surface of this flange, greatly appreciate your making it in this way instead of with the finished surface of the flange on the face side of the gear. It is much more difficult to measure accurately from the pitch line to the face side and at the same time hold the gear so that the pitch line of

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the teeth will run true. When the surface to be ground is on the back side of the gear, it is a simple matter to lay the gear face down on the grinding machine, supporting it so that it is located by pitch-line control, and then to produce a surface on the flange at the desired distance back of and in a plane true with the pitch line.

When an integral pinion, Fig. 16, is used, you must see that in the heat treating process the shaft is not sprung, or that in grinding the bearings each side of the pinion on the original centers, the gear does not run out of true. To get absolute accuracy and a high-grade product, the way to grind these bearings is to put the taper end of the shaft on a dead center and hold the other end—not on the shaft center—but by balls or rolls between the teeth of the pinion. The pitch-line of the pinion will then run true. Next grind the surface between the pinion and the taper end of the shaft. Afterwards the pinion can be reversed, putting the part already ground in a taper socket or collet on the grinding machine. The short end can then be ground concentric with the portion first finished. If this method is followed, the teeth of the pinion will run true with the axis of the pinion shaft and the gear will not make the intermittent and disagreeable noise so often heard in carelessly fitted gearsets.

J. G. PERRIN:—Axle builders grind bevel gears and spur gears by setting from the teeth. Several ingenious fixtures are used for holding bevels by the teeth, so as to grind the face and the inner bore of the gear true with the teeth. The same method is being used to a large extent in grinding transmission gears.

ADVANTAGES OF WORM GEARING

CORNELIUS T. MYERS:—One good way to get rid of the noise caused by the condition described by Mr. Heald, is to use worm gearing. It also provides a way of avoiding the eight- or nine-tooth pinions used to get a large enough gear reduction. Worm gearing permits any reasonable reduction, with the same road clearance given by bevel gearing. The teeth are so large and so strong that breakage is unknown. They can be ground after hardening to a greater accuracy than that heretofore thought possible. The worm is held to about a thousandth of an inch on the pitch line, and the variation in the regularity of the tooth surfaces is so small as to be difficult to measure with any ordinary instrument. Accurately cut worm gearing must be accurately aligned, in the first place, and it must be properly lubricated. Then, since it has been properly cut, the load will be carried on an oil film and not by metal-to-metal contact. There will be no noise; the efficiency will be high; and the gearing will outlive the car.

In evidence of the improvement in worm gearing I can state that five 1500-lb. delivery trucks owned by one company have been driven an average of 46,000 miles apiece up to November, 1915. The gear-

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ing was examined shortly after that time. It showed no wear and looked as though it had just come from the gear-cutting machines. All the trucks are still in service and have been averaging over 125 miles per day. The same results can be obtained with pleasure cars.

RESULTS WITH UNEQUALIZED REAR BRAKES

F. E. MOSKOVICS:—I want to ask Mr. Perrin about unequalized rear brakes. I have been driving a car for several months with unequalized brakes. The Packard Company also is using them.

J. G. PERRIN:—Many people are not using equalizers on the brakes now. If the brakes and the brake pull-rods are properly adjusted, they find no necessity for the equalizers. Future adjustment is left to the brakes themselves. This is certainly simpler and results in less weight and cost to the car manufacturer.

CHAIRMAN DUNHAM:—The objection to such brakes is in production. When the car is new, the brakes must be adjusted much more accurately than would be the case with an equalizer. One adjustment apparently does not do the work, because there may be a few high spots that will make one brake wear more rapidly than the other.

WHAT EFFECT HAS UNSPRUNG WEIGHT?

CORNELIUS T. MYERS:—I wonder if anyone here has any real data on the term that has been so bandied about for years, namely, "unsprung weight." What effect has it, and how important? When does it show itself, and where most strongly? Everyone talks gravely and mysteriously about "unsprung weight." Many arguments have been predicated on statements with reference to this term. Many of these statements have not been questioned because they seemed reasonable, but on what are they founded?

J. G. PERRIN:—The effect of unsprung weight is a broad subject. I think there is a happy medium between sprung and unsprung weight and if we go to either extreme, we lose in the best riding qualities of the car.

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MANUFACTURE OR ASSEMBLY OF PARTS

By H. M. Jewett

(Non-member)

Abstract

The question as to whether a part should be made or bought is one that must be settled by the individual maker according to the value of his product, the nature of the part, his capital available for manufacturing purposes and the price at which his product is sold. The author describes the practice followed by some of the large companies, showing that in spite of their being quantity producers, they have found it desirable to buy a number of important parts. Certain parts are rarely made by automobile manufacturers, either because they can be bought more cheaply or because the machinery to produce them is intricate.

The author sums up the problem by stating that a manufacturer makes the unit on account of not getting deliveries or because he does not get a fair price from the parts maker or an article good enough to satisfy his conditions. In order to give individuality to the product, the car maker often produces certain parts, such as the engine, himself. In conclusion, the paper pays a tribute to the standardization work conducted by the Society.

Should a manufacturer make car parts, or assemble them? What proportion of the car and what particular parts should the company manufacture? For the purpose of discussion, let us classify manufacturers as follows: First, those producing a high-priced car, selling for say three thousand dollars or over, in limited quantities; second, the volume producer, producing annually ten thousand cars or over; and third, the manufacturer that is forced into the manufacture of certain parts through business necessity.

The high-priced cars are produced in limited quantities. The very high-priced cars, over four thousand dollars, are produced in smaller quantities. In order to command such a price the car must have extreme individuality. The makers must really make practically everything in the car. A purchaser of a high-priced car must feel that every portion of it, the steel, the quality of workmanship, the limits required, are the best and the most exacting; that when he pays his money, he gets due value. The automobile manufacturer in the high-priced field can ill afford, I believe, to buy axles, engines, bodies, frames and wheels from manufacturers who are producing parts for cars in volume business. It would be difficult from a sales standpoint to convince prospective purchasers that individual parts of the car are any better than those made for the volume producer. The volume producer advertises that on account of his quantity he can buy more cheaply; so naturally the maker of the high-priced car, in limited quantity, has that argument against him.

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Some years ago one manufacturer of expensive cars sold two grades of body; one manufactured by a large body maker, and the other manufactured in his own establishment. The body manufactured by the specialist was sold at a price considerably lower than that of the body produced by the car maker. The home-manufactured body was worth the difference in price. That was the idea conveyed and the larger portion of the bodies sold, I will warrant, were home-manufactured.

PARTS-MAKING BY THE VOLUME PRODUCER

We will take as an example of the second class (producers of ten thousand or more cars annually) a maker of a hundred thousand units. It is commonly supposed that a manufacturer producing over ten thousand units, that is, a quantity manufacturer, can produce and manufacture parts more cheaply; that he can buy the material at less cost, lower his price, and keep within his sales possibilities by increasing his production.

Now, is this true? Take the largest manufacturing company, outside of the Ford, the Overland. It has been and is a large assembler. For years all cast-iron parts for Overland engines were cast and machined by a company in Cleveland. Various other parts, including bodies and axles, are bought. To-day in the case of the lowest priced car produced by this company, instead of casting the engine cylinders, and having tools and equipment to machine them, the castings are produced and machined by an outside company. There must be some good reason why the Overland Company has not gone into the manufacture of those units if it could produce them more cheaply and better. The Overland six engine is bought complete from an engine manufacturing company. I saw an axle housing in a plant in Philadelphia a short time ago. It was part of an order for fifty thousand for the Overland Company.

The Buick Company manufactures its engine. It buys axles from an inter-related company, of course. But its steering-wheels are purchased; also tires, certain accessories, lamps and a portion of the bodies.

The Studebaker Company purchases some of its gears, its frames, electric lighting and starting apparatus, batteries, and a great many other articles, including steering-wheels. Why does this company not manufacture these parts?

The Cadillac Company manufactures its engine and buys axles, purchasing, I understand, the steel for and manufacturing the gears, turning them over to the axle company. The Cadillac Company is a large volume producer. Why does it not manufacture all the parts of its car?

Taking the parts manufacturers, are they really manufacturers? Many of the axle companies buy all the gears used in their axles. Transmission manufacturers are buying gears and other parts. Many of the "parts manufacturers" are themselves assemblers.

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PARTS SELDOM MADE BY CAR MANUFACTURERS

There are certain parts that automobile manufacturers rarely make; anti-friction bearings, for example, possibly because they are patented and the machinery to produce them is intricate. Car-making companies can purchase them more cheaply from specialists, securing a better bearing.

A press suitable for making rear axle housings could produce all the housings needed, and lie idle practically three-quarters of the time.

A number of automobile companies have tried making steering-gears. It looks simple, but many who have started to manufacture them have gone back to purchasing them. A press to manufacture a frame for one of our largest car companies would not be profitable. It would produce the quantity in a few months and then lie idle; while the press working day in and day out on this, the other and the next man's product, will manufacture that frame much more cheaply. The result is, we buy and so do others. Speedometers, lamps, lighting and starting apparatus, and all the smaller accessories are manufactured by specialty manufacturers, and for a reason; they could not be produced in the car factory as cheaply.

ASSEMBLERS AND MANUFACTURERS

There are two kinds of "manufacturers"; one claims he is a pure assembler, the other that he is a manufacturer. A few years ago one manufacturer told the public that he was an assembler, pure and simple; that he purchased parts from the finest manufacturers in the country, who manufactured each unit of his car, and that his was the universal product. He believed in pure assembling. At the same time another plant advised that it had a corner in the chief engineers of the world and that those engineers could and did produce the highest quality individual units in his car. But the other manufacturer was telling the world that the only real way to produce a motor car is to assemble it. Both of those manufacturers are huge successes.

Walking through an assembling plant one day, I asked the superintendent, "Why do you enamel your fenders? You do not manufacture any other part of your car." "Well," he said, "We cannot purchase them good enough." Why are other people not enameling their fenders? Possibly because they have had good success in getting deliveries in enameling. That is the only answer, because, if the different manufacturers encountered trouble in getting deliveries, and in having enameling done properly, they would do just what this company did, put in their own enameling plant.

Going through various plants, what is found? Some will paint and trim. Go to another plant, of about the same capacity, and ask, "Why don't you paint and trim?" "I would not paint and trim

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for anything in the world; what is the good of it? I can get just as good a job done here at Brigg's, the American Auto Trim, or somewhere else; I don't have to use the floor space; I can get it done more cheaply." That is his idea from experience in the business; and he acts accordingly.

In other plants every part of the engine is bought and assembled, because the engine manufacturer did not want to build a special engine at the right figure. There were perhaps other reasons peculiar to given companies. Instead of putting money into machinery, tools and equipment, the parts are manufactured outside, brought in and assembled. Each quantity manufacturer has a different condition to deal with, and approaches it in a different way; and each one of those ways is the result of personal experience in the business.

CAR MAKERS PRODUCING PARTS

Generally a car manufacturer makes a unit on account of not getting deliveries, or a fair price from the parts maker, or a good enough article to satisfy his conditions. It may be that he does not buy a standard engine, because it is used by several different car manufacturers with products of various prices. He may believe that his engineering staff has designed an engine that is the logical one for his use, under his conditions. So with each one of the different units of the car. A man manufactures or buys according to conditions that arise from time to time. I think that it is better for the car maker to purchase the majority of the engine parts, as they can be manufactured as well by the parts maker as by the car maker.

At one time we manufactured our own engine. The engine was designed in our engineering department. The only way we could get it was to build it. It was a four-cylinder engine. The time came when sales conditions demanded a six; so we bought an engine. When we see a sales possibility in a certain field, we cannot wait until it is gone; we must produce the car, buying the parts.

CAPITAL REQUIRED PREVENTS PARTS MAKING

Think of the capital investment an automobile company producing 10,000 cars a year would be put to to install the machinery and equipment, buy the material, manufacture it and turn out a finished product. Think of the millions of dollars it would take. But when the company purchases the different parts from accessory or parts-making plants, their capital is being used in the business; and just as long as car companies can buy those parts of as good quality as they think they ought to be, and at as good a price as they feel they should get, not many automobile manufacturers will go into the parts-making business.

The engine is the most talked-of part of the car. There is the individuality that the car maker would like to have in his engine.

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There are good reasons for feeling that the engine is the most logical thing for an automobile manufacturer to produce, aside, of course, from little accessories, such as pedals, which it is easy to farm out, but which cannot be produced outside at a figure at which they can be at the home plant. It is the small articles, I find, that cost the most money relatively.

RESULTS OF STANDARDIZATION

The Society of Automobile Engineers has made possible the standardization of the motor-car industry, which is an infant, but also an intricate business. To-day a chief engineer of a plant can look over a blueprint of an engine and tell whether the crankshaft is of proper dimensions, and the bearings are right. He can tell whether the steel in a transmission or in various parts of a car is able to hold the horsepower of the engine. I think that the automobile manufacturers owe to the engineering brains of this country a debt they cannot pay. There is a great and crying need for transportation in this country. The engineers have simplified the manufacture of the automobile, and too much praise cannot be given them.

DISCUSSION

CHAIRMAN D. MCCALL WHITE:—In one particular price class automobile manufacturers make many of their own parts in order to obtain a special individuality which they cannot possibly obtain by buying parts that someone else uses. Parts are also made to obtain refinement of workmanship. Then there is the quality of material, which is connected with durability. Some cars have a reputation for durability that in itself raises their second-hand value. One of the great selling arguments is the second-hand price.

The makers in the quantity-production class in which parts are bought have trouble with lack of delivery. Sometimes practically only one accessory manufacturer makes a certain part. This company may have so many orders that it is absolutely incapable of producing parts with special refinements, or in sufficient quantity to meet requirements.

The price that some of the accessory or parts manufacturers charge often enters into the question. One company, by reason of special equipment, may be able to manufacture a particular part more cheaply and better than the parts maker can. On account of lack of floor space and sometimes for financial reasons, companies cannot install the special equipment necessary.

REASONS FOR MAKING PARTS

ALVAN MCCAULEY:—We established a policy of manufacturing our own parts many years ago, and yet we have proceeded on the policy that whenever we could purchase the kind of part we needed, of the quality that had to go into the car, and that we were putting

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into the rest of the car, we would not hesitate to go out into the market and have that part made. We have made a number of ventures into the purchasing of assembled units; sometimes with gratifying results, sometimes without. When the Packard Company started there were few, if any, reliable manufacturers of parts; so we were driven to manufacture all of the car, and early embarked on the basis that we would re-invest our earnings from year to year in our plant, until we should have the most complete plant in the world.

It was not long before Mr. Ford and others outstripped us in that respect, and showed us the folly of some of our reasoning. But we like to impress our own individuality on the various units of the Packard car, and so to-day prefer to make them ourselves. We feel almost that when a manufacturer does not make his own parts, it is because he prefers to have the liquid dividends rather than re-invest his earnings in expensive buildings and equipment. It takes a gigantic investment to make your own parts.

The tendency of the times is bound to be more and more toward manufacturing of parts and units by the car producers themselves. It seems to me that the dividing line is drawn along quantity manufacturing. There are some modifications to that. Willys, although a big manufacturer, does not produce all his parts, but the reason is that he has not had time to produce them; his business has increased by such immense leaps and bounds that he could not erect buildings fast enough, or get machinery in sufficient quantity. He had to buy. He is, however, making more and more parts; he is acquiring interest in tire plants, and owns his own engine starter business, his own foundry and forging plant. He will find in the end that he will make everything he possibly can. As a manufacturer increases his output to the point where he can himself compete in price and quality with the units that he can buy outside, he will gradually produce those units.

CORNELIUS T. MEYERS:—We are confronted with the tendency in the automobile industry to cover up or gloss over what we are doing, mostly to our own disadvantage. A little plain talk throws light on the rough spots and clears our minds for better efforts. A company first uses one construction; then turns around and uses another. The change was made because conditions had changed. We, as engineers, have a commercial purpose to serve. Unless we design parts that can be secured in sufficient quantity and of proper quality, manufacturing cannot go on. Sometimes we can purchase these parts, sometimes we call on our own plants to make them. We also find other arguments in favor of this particular construction or that particular "improvement," but at bottom it is often a question of expediency, of engineering design and compromise. Every one wants to produce the best possible; despite the fact that every one wants to make a profit, he has really great pride in his

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product, and is looking continually to see how he can get this or that part which will suit his purpose better and assure more dependable and efficient performance. Whatever we use we buy, paying either another organization or a part of our own—preferably the former, in almost every case, because of the relief from the buying responsibility for detail and investment; it is only when the outside source of supply fails to measure up to our engineering and commercial requirements that we resort to the latter. The reason many manufacturers in every price class are forced to manufacture for themselves, is to have a more reliable product; that, I think, is fundamental.

ADVANTAGES IN BUYING PARTS

E. P. HAMMOND:—The company I represent makes steering-gears. Over 75 per cent of our business to-day is with companies that at some time manufactured their own steering-gear. One of our largest customers is a company that, when it first started, bought steering-gears and later made them. Three or four years ago we convinced that company that we could make a better and cheaper product. But I do not say that at some future date that company will not manufacture gears again; men's minds and conditions change. The general tendency to-day—and this has been true for the last three or four years—is that car manufacturers are giving up making steering-gears.

One of the advantages an automobile company has in buying parts from a specialist, is that the organization of that specialist is thinking of nothing else than what he is making. His engineering department comes into contact with all the engineers in the industry; is constantly picking up new ideas, and putting them into the product. It seems to me that this intercourse alone is sufficient to show that the specialist can develop as good if not a better product than the man who is making the engine and transmission and putting the car together. The specialist has to meet competition from the independent parts companies, as well as from the automobile manufacturer who is making parts; he is constantly endeavoring to keep the latter from becoming interested in manufacturing his own parts. The specialist is developing his design toward quality and economy; he can put a larger investment into tools to make a particular part. I have known of several cases in which he has been able to buy more cheaply because he had standardized his product and was thus able to place orders for large quantities of a given material.

INCENTIVE TO MAKE PARTS

K. W. ZIMMERSCHIED:—Mr. Jewett has treated the subject from the business viewpoint—a viewpoint of great value to engineers. Engineers have a strong preference for diagrams and curves, and perhaps I can best present my ideas with a chart.

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Automobile factories can be divided, first, according to the quantity of cars produced, and second, according to the price per unit. I would put into the first quantity class the plants that produce up to 10,000 cars per year; into the second those that produce from 10,000 to 30,000, constituting what might be called medium-quantity-production factories; in the third those producing 30,000 to 100,000 cars per annum, constituting a large-production class; and in the final class those producing over 100,000 per year—constituting what might be called a super-quantity-production class.

Considering price, we have a class of cars selling for $3,000 and up; a second ranging between $2,000 and $3,000 per unit; a third

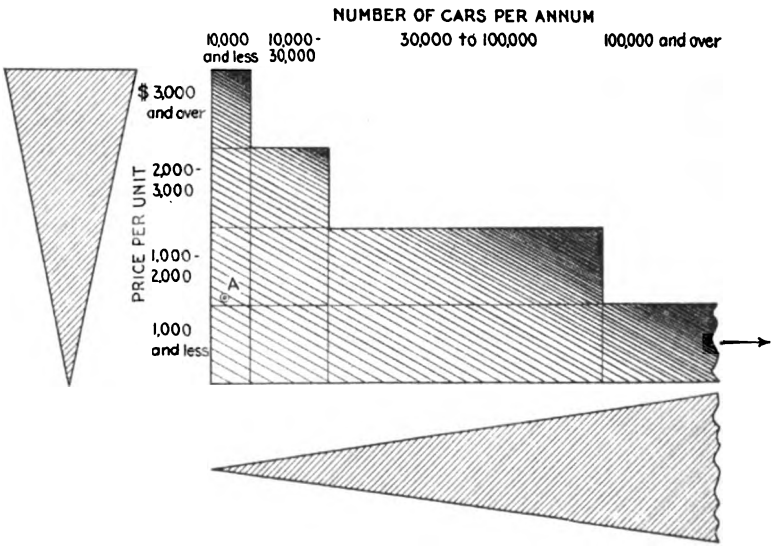

CHART SHOWING INCENTIVE OF MANUFACTURER TO MAKE PARTS

comprising those sold for from $1,000 to $2,000; and a fourth class comprising all cars sold for less than $1,000. Such a classification cannot be filled out, because the production class runs into outputs that no company in the higher-price classes has ever been able to market. The accompanying diagram indicates, in a rough way, those portions of the field that are covered in the automobile industry as it stands to-day.

The horizontal and vertical wedges indicate at any point along the scale, for production and price classes respectively, the incentive a manufacturer has to manufacture his own parts, the width of the wedge representing, according to some mental scale, the magnitude

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of this incentive. The manufacturer of expensive cars is called upon to produce most of his parts because his product must incorporate extreme individuality. As the price decreases, the demand for this individuality of appearance, workmanship, and general engineering excellence decreases until, within the lowest class, there is the minimum incentive for individual production.

Considering the matter of quantity alone, those manufacturers having the smallest annual output have the least incentive to manufacture parts, but as quantity increases, it becomes increasingly profitable to manufacture the individual parts and we therefore find the maximum incentive in the super-quantity-production class.

The situation existing in any of the areas on the diagram can therefore roughly be pictured as the resultant of these two wedges, and on this basis it is probable that in the low-price, low-quantity production area in the lower left-hand corner the cars would be produced wholly from assembled parts. On the other hand, either the high-price class or the super-quantity-production class at the extreme ends of the diagram would, to the maximum degree, be made from parts manufactured by the car builder.

Supposing that a group of individuals intend to enter the automobile manufacturing field, they would probably take the line of least resistance and start with a car that would supply a steady and conventional demand, making no pretense to produce exceptional quantities at first. The logical position of such a product would be at about the point indicated by *A*, the car itself being manufactured almost entirely from assembled parts. But after a year or two, such a company would find its sales department playing an up-hill game—hampered by inability to offer individuality to customers, it would probably be impossible to market an increasing number of cars each year. The manufacturers would therefore endeavor to find some means of expressing greater individuality in the product. This individuality could be expressed in two ways—by individuality in appearance or of performance; because the motor car has a dual function, a utilitarian one as a means of transportation, and a personal-adornment function as a means of advertising the owner's wealth.

ENGINE INCREASES INDIVIDUALITY OF PRODUCT

The most logical unit to pick out for increasing individuality of performance is the engine. But the car builder finds that he cannot obtain a product of individual characteristics from standard engine builders at a reasonable price, because the conditions that make for low price disappear when the product is made in the small quantities he can absorb. The car manufacturer, therefore, must either build his own engine or take a sufficient quantity to make his business attractive to the engine builder. If he decides to increase the quantity, the reasons for his producing the engine himself are

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again strengthened. If he secures tools for this, he probably finds a number of other parts about the car that he can also manufacture to advantage.

The success of the manufacturer in these departures depends upon the plant equipment and the engineering ability at his disposal. If he can keep the equipment busy, thus holding down overhead costs from this source, and if his engineering talent is equal to the task of designing and producing a creditable product, he will secure the results desired. If, however, either of these essentials is lacking, he will either produce an inferior product or his cost of manufacturing will exceed expectation. Unless the analysis of costs is correct the latter condition might obtain for some time before the real cause for loss would appear.

One of the greatest advantages of buying parts from outside sources has to do with the matter of engineering talent; it is too much to expect such an engineer as could be hired for what a small plant can pay, to have the experience and the ability to design and produce all the parts of a car of a quality equal to that of the product of companies specializing on each unit. A staff of engineers, each with his province, is even more expensive. We owe a tribute to the parts manufacturers for the great advances many of them have made in their respective fields. Parts manufacturers competing for one another's business spur their engineering departments on to produce units of equal quality at a lower cost, or of better quality at the same cost.

We all appreciate Mr. Jewett's tribute to the automobile engineering profession and feel that this should be shared with the engineers of parts manufacturers, who have occupied so prominent a place in the industry and in the standardization work of the Society of Automobile Engineers.

J. W. WRIGHT:—We make a specialty of engines, doing assembly work mostly. At one time we did our own machine work; in other words, manufactured our own engine. But we found that was not a paying proposition. In order to manufacture in quantity we had to add an immense amount of equipment. We have been benefited greatly by placing orders for engine parts with manufacturers who make a specialty of them. We are not subject to as great a loss in manufacturing. It is sometimes thought that in getting parts finished outside we are paying an extra profit that would more than make up for the loss from parts spoiled, but we have not found this to be the case. We would not think of again manufacturing all our parts. Those who are manufacturing in large quantities and can put in equipment enabling them to compete with the parts manufacturer, might take a different view. Our largest customers are car manufacturers who at one time made their own engine. To some of these customers we are furnishing engines in lots of ten thousand.

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DIFFICULTIES IN BUYING PARTS

J. G. VINCENT:—I would like to speak of the engineering phase of this question, leaving out the commercial side. We will assume that an engineer has designed a car and tried it out and is thoroughly sure that everything is right. Suppose that the company wants to buy the units of that car. Unless the production is extremely large and the company can place orders that will occupy a large portion of the parts manufacturers' facilities, the latter will not consider seriously making the units. They will supply some standard product. The line of least resistance is, of course, to adopt the standard units.

After the manufacture of the car is started, in order to get the best product, a competent inspector is sent to the parts plants, and the trouble begins. Gradually prices will begin to rise, because the usual product of the parts manufacturers is not passed by the inspector. The first thing the management knows, the price of the various units has risen considerably.

The parts manufacturers have, I believe, in many cases stepped a little beyond reasonable bounds in order to get business, and after a short time, when they are held a little too closely on limits, stopped filling orders. What is the car manufacturer going to do then? A new set of drawings must be prepared immediately. The old units are obsolete, so far as the factory producing the car is concerned, but in some way service parts must be secured. Service is one of the hardest things to handle in connection with the assembled car.

The real advances in the industry have been made by those making their own units. Good work has been done by the parts manufacturers, particularly in producing good parts cheaply. But quantity is required, of course, for cheapness. If a manufacturer is producing enough cars, and has sufficient capital, he can produce just as cheaply. On the other hand, with a high-priced car, he produces the parts at whatever cost he can. The engineer in charge of design for the company certainly has the best chance to study the construction scientifically, and to change a part here or there, or the whole engine, or the transmission, when he has really developed something better.

There are many parts which I think no automobile company will ever manufacture; but I rather draw the line at parts that are not absolutely standard. If one company becomes tired of selling ball bearings, they can be bought from somebody else, without changing the design, or causing a bad service condition. I believe thoroughly in buying such parts, providing they can be bought satisfactorily.

STATUS IN PERSONAL ELEMENT FIELD

GEORGE W. DUNHAM:—Let us consider this subject in a somewhat different way, dividing cars into three classes; first—that of extreme quality; second—a personal-element class, and third—a class of extreme quantity.

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The first and third classes have been covered thoroughly in the discussion, and it has been shown clearly that their makers should, and actual practice develops the fact that they do manufacture most of the car parts. The personal-element class includes cars made by the majority of manufacturers; there are in it more manufacturers than in the two other classes taken together. I refer to the manufacture in quantities up to 20,000 or 25,000 cars per year, ranging in price from about $700 or $800 up to $1500 or $1600. Such manufacture usually starts with an assembled car. Because of difficulty in getting suitable prices or proper deliveries of parts, or the quality desired, the companies have taken up the manufacture of the different units—perhaps one at a time, but nevertheless, they start in the manufacturing business.

In this middle-price middle-quantity class, whether parts shall be manufactured or purchased depends largely on the personal element, that is, on the feeling and opinion of the men in charge of the company. If the man at the head of the factory understands the machine shop thoroughly, he will undoubtedly favor building at least some of the parts of the car. Financial conditions affect the matter strongly; to manufacture it is necessary to invest much money in machine tools and equipment. Excellent financial resources are necessary also to carry the heavy burden of the large amount of material required.

OTHER SIDE OF ROAD LOOKS BEST

The first thing to be impressed on a new company entering this middle-class field is the old saying, "The other side of the road always looks the best." If companies are manufacturing their own parts, many times they wish they were not. They will see someone else buying everything, and feel that he is simply buying parts and putting them together, the parts-maker carrying the burden. Then again, if they buy all the parts, they run into difficulties—engine trouble, or some trouble with transmissions, axles or other parts; and they immediately decide that if they were manufacturing the particular part themselves they would have avoided all difficulty. Trouble exists no matter which way you do it. Up to a certain point, funds being available, of course, the personal equation is the deciding factor. One naturally takes great pride in having a large well-equipped machine shop. This very human feeling, plus the selling advantage gained, has no doubt influenced more than one of the car assemblers to manufacture parts

The ideal way to build a medium-priced medium-quantity automobile, from the money-making standpoint, is to assemble it, and were it not for the personal element of the parts-maker, many car manufacturers now owning large machine shops would still be assemblers. It is probably good business for the parts manufacturer to sell his product to the best advantage, and to the customer who is

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the least particular as to quality, but if I were making motor-car parts, I would make them so good and on so close a margin and give such good service, that my customer would not think of manufacturing himself. I would try to do so well by him that he could not afford to manufacture. It would be my aim to build up and increase my market, rather than tear it down, as now seems to be the intent in many cases. Competition is forcing the manufacturer of automobiles, especially in this personal-element class, to give greater quality at a less price. Some of the parts manufacturers are making a grave mistake when they do not hold the connections already made, and do their part as to quality and price so satisfactorily that the car manufacturer will continue business with them year after year. Just so sure as the present average conditions continue to exist, will the automobile builders take up the manufacture of their own parts, gradually eliminating the market for finished units.

T. P. CHASE:—The history of the King Company will illustrate many of the points that have been brought out in the discussion. We started about four years ago to build a popular-priced car. We bought all the parts and, mainly on account of limited facilities for manufacturing, it was necessary to buy assembled units. After a while we found that on account of different conditions we had to either change the source or do something to get engines. Not wishing to change the design of the engine all the way through, on account of service, we decided to build it ourselves. We did so and went along fairly well, until we found we were in a class in which it was hard to get volume. Then we conceived the idea, soon after the Cadillac started with the eight, that we could get a little individuality in the car, and that it might be a good idea to bring out a popular-priced eight; so we designed our own engine. We had considerable difficulty in buying an assembled engine of that type and of necessity again built our own engine, building our transmission and buying the rest of the parts.

In a little book published about a year ago by the *Saturday Evening Post* it was said that the automobile is two things—a machine and a style carriage. To get individuality in style the car-maker must design the body lines and the general lines of the car, and that body must be built to that design without much trouble. It is difficult to buy an engine with any individuality from the parts manufacturers. In order to get volume the parts-maker has to build engines that other companies want, and individuality is lost.

INCREASE IN DEMAND PREVENTS PARTS-MAKING

E. V. RIPPINGILLE:—Perhaps one of the reasons why more car companies have not gone into the manufacture of parts is that the demand for their product has grown faster than they could put in the machinery. I believe that most of the lower-priced car manufacturers find themselves in this position; sometimes they form sub-

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sidiary parts companies and afterwards absorb them. Several companies have taken up the manufacture of one unit at a time.

I believe we like to *think* we can do things better than anybody else. The specialist has helped a great deal, and will, I believe, always dominate in some lines. Nevertheless, in the case of certain units of an automobile individuality and the reputation of the manufacturer must be "built in." So the manufacturer finds that as the need for individuality and better workmanship grows, he must give it his personal attention. The experience of the Hudson Company has been along those lines. We have put in one piece of equipment after another, just as we have always built a few more cars each year. This has been done from a protective standpoint, for two reasons: We must protect ourselves financially and as to individuality and workmanship.

The specialist has undoubtedly helped the car manufacturer in determining what he should make himself. There will always be specialists in every line, because automobile individuality is far from being fully developed.

W. A. BRUSH:—I believe that the parts-makers, who are in a way specialists, have not aroused themselves to what the word "specialist" really means. The specialist, if he is a true specialist, must produce something better than anyone else in his line can. The specialist in parts-making to-day assumes as a rule that an increase in quantity means an increase in dividends, but not an increase in quality. If he would become a true specialist, the increase in quantity would carry with it improvement in quality. Then the automobile manufacturer would have less reason to make his own parts, and more reason to buy parts from the parts-maker.

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ENGINES OF THE AIR

By J. G. Vincent

(Member of the Society)

Abstract

After a brief consideration of airplane-engine practice in France, England and Germany, the author outlines the problems encountered in designing a twelve-cylinder aviation engine. He explains at some length the difficulties in determining the connection between propeller and engine and shows why valve-in-head location was chosen. Such features of engine design as the mounting of carbureter and exhaust pipes, methods of fuel and lubricant supply and details involved in selecting the lighting, starting and ignition equipment are considered.

In considering the subject of aircraft engines, the questions arise: What is the best type of engine and what are the best details of its construction? A considerable difference in opinion existed in Europe at the beginning of the war, as to these matters.

When the Germans started active work on their air equipment they asked automobile engineers to develop proper engines. The result was naturally a high-efficiency engine of the automobile type. The Mercedes is probably the best known type of German engine; it is nothing more or less than a vertical six-cylinder engine, of very light yet reliable construction.

AIR-COOLED ROTARY ENGINES FAVORED IN FRANCE

In France the engine designed was handled by aircraft engineers. The strongest tendency was toward air-cooled rotary types. These were developed to a high state of efficiency, and a remarkable power per pound of weight was obtained from them. During the trials, however, the engineers did not run any long flights, or take into consideration the weight of fuel and oil to run the engines. When the war came on the scouts were required to make long flights and it was dangerous for them to land in the wrong place. It was then found that the aeroplanes had to be loaded with so much extra fuel and oil that the working weights of the air-cooled rotary types of engine were greater than those of the automobile engines with water-jackets.

The condition was slightly different in England, where an aircraft engine had been developed more nearly along conventional automobile lines.

FIXED-CYLINDER ENGINES BEST FOR WAR SERVICE

At the breaking out of the war, or shortly afterward, the French were forced to adopt new principles. They developed water-cooled

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engines of the fixed-cylinder type. Some companies followed the German practice, using six-cylinder water-cooled engines. Others developed air-cooled V-type engines. There is still some difference of opinion, of course, as to which is the best type. From what I can learn the rotary type of air-cooled engine has a place in light scout machines, where extreme speed, which means light weight and small plane surface is desired. But for carrying observers or ammunition for long-sustained flights where extreme reliability is desired, that type of engine seems to have been replaced by the fixed-cylinder type.

I do not believe that any one of the fixed-cylinder types is absolutely essential. The type selected should depend on the power desired. However, larger planes, carrying more passengers or ammunition, are wanted and therefore more power is required. There is a limit to the six-cylinder bore and stroke, so that a V-type engine of some sort will very likely be used.

The next step would be the V-type eight, which has been built and been successful. But the cry has been for more power. The twelve-cylinder V-type engine is favored and a movement is on foot to build eighteen cylinders, that is, three rows of six cylinders. That is practicable, but has the disadvantage of being complicated, so far as connecting-rods are concerned. Its width is a distinct detriment to fast flying.

The eight also has the disadvantage of being too wide. The engine should be as narrow as possible. A body for high speed should not be over 26 in. wide over all. It is difficult to keep a large engine, even a 60-deg. unit, inside of those dimensions. I would, therefore, say that for large plane work nothing less than twelve cylinders will probably endure. The rotary air-cooled engine will probably be used for scout work to some extent.

With this information before us, and with a feeling that some one in this country should develop aircraft engines, partly because we believe they may have a commercial future, and also because we think this country may need them, we have developed two twelve-cylinder aviation engines.

DIRECT OR GEARED PROPELLER CONNECTION

The first important detail discussed was whether the propeller should be connected directly or through gears to the crankshaft. If it is mounted on the crankshaft the engine is limited to a speed of from 1200 to 1400 r.p.m., and can be used only in light fast machines. In the heavy load-carrying machines it is sometimes desirable to run the propeller as low as 900 r.p.m., so that it seemed desirable to gear down the propeller. Many difficult problems are involved in the geared-down construction. The direct-driven machine, with propeller mounted on the crankshaft, is not altogether free from troubles. The propeller will be unbalanced in spite of fine workmanship. This condition, coupled with the inequalities of the air, puts some heavy strains on the propeller and its mounting. Some of the foreign governments are testing propellers by mounting a 7-lb. weight 30 in.

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from the center of the propeller, and running it at its rated speed for a considerable time. That may not sound like much of a test, but it will nearly lift a dynamometer from its base at 1400 r.p.m.

The unbalanced condition in propellers is, I believe, responsible for crankshaft breakage on the direct-driven aeroplanes. The result is that the crankshaft is one of the weak links in aeroplane-engine construction.

In using the geared type the strains due to the propeller are absent on the crankshaft, but are taken by a short lay-shaft. This must be rigidly mounted on heavy bearings. Such bearings must provide for radial load and also for end-thrust, so that the use of large self-aligning ball bearings at both ends of the short shaft and a heavy double-thrust bearing in between seems to be the best practice. Substantial gears are then required, and ample means must be provided for cooling them with oil. I believe that with a proper design an efficiency of between 98 and 99 per cent can be obtained with the gear drive. The crankcase and the case to carry the propeller mounting should be cast of an alloy a little stronger than the ordinary aluminum.

PROPER LOCATION OF VALVES

The next thing to be considered is the location of the valves. We want the most power, coupled with the best possible economy and reliability. Noise is not so much a factor, so that the valve-in-head construction is naturally the choice. Two overhead camshafts are not absolutely necessary but seem desirable. They do not add greatly to the weight of the engine, but do decrease that of the reciprocating valve parts greatly. This is particularly desirable with the geared-down propeller, because the engine will probably be run in the neighborhood of 1800 to 2200 r.p.m. In an aeroplane the engine runs always at its rated load. It is not like an automobile engine that is working at about one-tenth load the greater part of the time.

MOUNTING OF CARBURETER AND EXHAUST PIPES

The overhead camshaft is of clean-cut construction, and permits a good clearance for the carbureter and the exhaust pipes. There are two general schemes for mounting these. One is to put the carbureters on the outside and the exhaust pipes in the center. The other is to mount the carbureters in the center, and the exhaust pipes on the outside. The second method is desirable because the spark-plugs should be on the inside, where the oil will drip off them. It would be undesirable to have both the exhaust and the spark-plugs on the inside, as pre-ignition might result.

TYPES OF CYLINDER DESIGN

A number of cylinder designs are fairly satisfactory. The straight cast-iron cylinder can be made of light weight, especially when the jackets are welded on. Another construction is to use aluminum cylin-

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ders with steel liners, either casting the cylinder separately with the head integral with the cylinders (the cylinder must then be removed to take a valve out), or casting the main barrels of the cylinder integral with the crankcase, pressing in the liners, and bolting on the heads. The aluminum cylinder is better than the cast iron on account of the greater weight of the latter.

The Mercedes cylinders are forged of steel. In my opinion, the steel cylinder is the best, although it is the most expensive, and the hardest to make. Considerable experimental work may be required before we find how to make them as well as they do in Germany.

LIGHTING, STARTING AND IGNITION

We consulted with a number of foreign aeroplane engineers, as well as with our American engineers. Practically all of them wanted electric lighting and starting apparatus installed if it could be done so that it would operate successfully. Electric lighting and starting seem to be needed just as much as and even more than they were in the automobile. If the engine stops in the air it is desirable to be able to start it. We are therefore trying to secure the best possible results from the electric lighting generator and starter motor.

The foreign engineers wanted two spark-plugs more for reliability than because they were necessary for power. They also wanted magnetos because they were not familiar with the present generator ignition in this country. An aeroplane engine should be designed so that either magnetos or the generator type of ignition can be used. As electric lighting and starting become popular, as I am sure they will, the generator type of ignition will go with them.

SUPPLY FOR FUEL AND LUBRICANT

It is necessary to have an air-pump for providing either pressure or vacuum. The tendency seems to be growing to use vacuum-feed for the gasoline on account of the liability of holes being shot in the tanks, thus releasing the pressure.

Lubrication is of the pressure-feed type. Two general systems are in use—one is to carry the oil in the sump, and the other is to pump it all out of the sump, and through a radiator.

WORKMANSHIP FOR AVIATION ENGINES

The aviation engine will require workmanship that we could not possibly afford to put in an automobile, and it would not do any particular good in an automobile. It is almost impossible to put a gasket in an aeroplane engine. It might be used between the intake and the carbureter, but nowhere else. The constant weaving of the engine in the air seems to work the gaskets loose and they blow out. Every joint must be lapped, so that it can be taken down any number of times without leaking.

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INDIANA SECTION PAPERS

INCREASING AUTOMOBILE-ENGINE THERMAL EFFICIENCY

By C. E. Sargent

(Member of the Society)

Abstract

The efficiency of the automobile engine, as operated on the Otto cycle, is thought by the author to be too low, and he therefore suggests a method of improving it. He considers the various losses by which heat is dissipated in internal-combustion engines and finds that the best opportunity for increasing thermal efficiency is by an increased expansion of the charge. The author suggests that this expansion be carried 50 per cent further than is ordinarily done.

In order to obtain higher efficiency at part load, the suggestion is made that instead of throttling the mixture, the admission valves be closed earlier. In case the expansion is 50 per cent longer than the induction stroke and the cut-off takes place earlier as the load becomes lighter, it will be necessary to vary the fuel opening inversely with the air induced. It is suggested that the fuel valve and cut-off lever be connected together and operated by the accelerator pedal or hand lever on the steering wheel.

The object of this paper is not to criticise the thermodynamic cycle of the present automobile engine, but to point out its inherent inefficiency as operated, and to suggest a method of converting more of the heat of the fuel into useful work than is ordinarily done.

In no industry has development been more rapid than in the manufacture of automobiles and their accessories. Driven by a prime mover, the prototype of which weighed from 300 to 400 lb. per brake horsepower, the complete automobile to-day seldom exceeds 100 lb. per horsepower of output. Improvements in carbureters, ignition and oiling systems and the discovery of alloy steels—these, together with careful attention to details, have made reliability an attribute of both engine and car. As reliability became an assured condition, attention was given to other things, such as quiet running, engine capacity, accessibility of parts, and finally to the turning moment, which has resulted in the multiplicity of cylinders found in many cars to-day.

Before the automobile arrived, gasoline was a drug on the market. The price was correspondingly low, so that the fuel consumption of the early engine was given but a passing thought. But with the specific gravity and surface tension of fuels increasing, and the price 400 per cent higher than in the early days, one of the greatest present needs in the industry is cheaper fuel, or more power from the same quantity of fuel.

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THERMAL AND MECHANICAL EFFICIENCY

The thermal efficiency is equal to the heat converted into useful work divided by the total heat supplied in the fuel. In Fig. 1, which is characteristic of internal-combustion engines, of the heat supplied 20 per cent is converted into work and 80 per cent lost or rejected in friction, cooling-water radiation and exhaust. To convert more heat into work one or more of the losses indicated must be decreased.

With good workmanship, force-feed oiling and light reciprocating parts the mechanical efficiency (brake horsepower divided by indicated horsepower) is about as high as can be expected, and we can look for no further appreciable gain from that source; as a matter of fact, only 5 per cent of the total heat supplied is lost as engine friction.

FACTORS IN WATER-JACKET LOSS

The loss of heat to the water-jacket depends upon the difference in temperature between the burning gases and the cylinder walls, the time the cylinder walls are exposed to the heat of the flame and the amount of surface exposed. If we could maintain the combustion chamber at the same temperature as the inclosed gases there would be no difference in temperature and no transmission of heat, consequently no loss to the walls and water-jacket. Such a temperature, however, would not only destroy all vestige of lubricant but would also melt the cylinder walls and pistons.

The time per working stroke in which the difference of temperature exists is decreased by high speed, wherein lies the advantage of high piston speed in an internal-combustion engine. The thermal efficiency

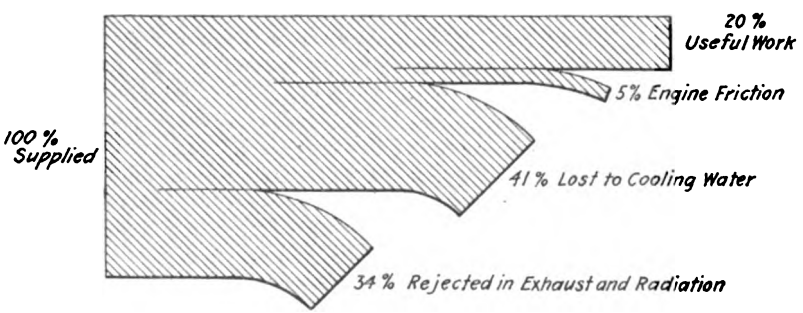

FIG. 1—LOSSES IN INTERNAL-COMBUSTION ENGINES

of big guns lies in the missile speed—nearly 3000 ft. per second—while the maximum piston speed obtained in an engine cylinder probably never exceeds 40, and seldom averages over 20 ft. per second. One of the most economical internal-combustion engines ever designed had a free piston, which, returning to the cylinder by gravity, converted more heat units into work than has any engine with a restricted piston.

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The requirement of a minimum value of the ratio of the explosion-chamber volume and surface is one reason for placing valves in the head. The advantages of multicylinder engines with their high speed, light reciprocating parts and minimum angular-velocity variation seem to offset any thermal losses due to increased surface per unit of volume, a characteristic inherent in such engines.

Even with a spherical explosion-chamber, the ratio of the surface to the number of heat units within decreases with compression, therefore the higher the compression the less the cooling surface exposed per unit of heat liberated. The higher limit of compression pressure is, of course, premature ignition, a condition incompatible with successful operation.

The possibilities of increasing the thermal efficiency of automobile engines by decreasing the jacket losses lie then in a high piston speed, a minimum surface per unit of volume (characteristic of the sleeve-valve and the valve-in-head engines), and as high a compression as possible without spontaneous ignition.

EXHAUST LOSSES FROM EARLY VALVE OPENING

Small gains in efficiency have been made by reducing the mechanical friction and decreasing the losses to the water-jacket, but the loss of heat to the exhaust has had but little consideration.

Several compound engines of the internal-combustion type have had a mushroom existence, but when we consider the added surface of a larger cylinder and how quickly the pressure after ignition becomes less than the pressure of compression, we see the improbability of reducing the losses by this means.

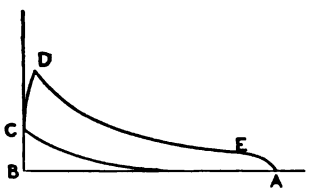

FIG. 2 — ORDINARY FULL-LOAD DIAGRAM

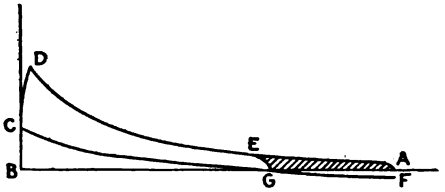

FIG. 3—FULL-LOAD DIAGRAM WITH LONG EXPANSION STROKE

It is necessary to open the exhaust valve from 40 to 60 deg. before the end of the working stroke in order to get rid of the back pressure on the piston during the exhaust stroke. The release of this exhaust, at 30 to 40 lb. absolute pressure, with its accompanying "bark," the visible flame and red-hot exhaust manifold are ample evidence of thermal inefficiency, even without the proof deduced by a heat balance.

MORE WORK WITH INCREASED EXPANSION

Let Fig. 2 represent an indicator diagram of an internal-combustion engine developing its full power, in which *AB* is the piston stroke and atmospheric line; *C* the highest compression; *CD* the firing

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line; *DE* the expansion line and *E* the exhaust opening. A cylinder full of mixture (*AB*) is compressed to *C* and ignited, raising the pressure to *D*, and as the piston moves down the pressure drops to *E*, where the exhaust opens.

By opening the exhaust at *E*, Fig. 2, the volume of hot gas released is less than that of the cold mixture at the beginning of compression. If, instead of opening the exhaust at *E*, we carry the expansion of the burning charge 50 per cent further, releasing the gases at a lower pressure, at a lower temperature and at a point nearer the end of the working stroke, more heat will be turned into work, because less will be rejected in exhaust, less power will be required to open the exhaust valves, and the muffler will not be required.

Fig. 3 shows a full-load diagram in which the expansion is 50 per cent longer than the compression stroke, *AB* representing the length of stroke and atmospheric line. The piston, starting at *B*, the beginning of stroke, draws in a charge to *G*, when admission ends. The absolute pressure in the cylinder drops to *F* when the piston reaches the end of the stroke; *FC* represents the compression line, which crosses the atmospheric line at *G*. The compression pressure *BC*, the firing line *CD*, and the expansion line from *D* to E would not vary from those of the ordinary engine, provided the mixture and timing were the same and the clearance was the same as an engine having a stroke *BG*.

The shaded portion of the diagram *EAG* represents the power gained, using the same amount of fuel, with an expansion stroke 50 per cent longer than the induction stroke. The shaded portion averages 16 to 20 per cent of the whole diagram, indicating a 20 to 25 per cent increase in thermal efficiency, or a corresponding increase in power from the same amount of fuel. An engine using such a cycle may weigh a little more per horsepower, just as a long-stroke engine is heavier than one with the stroke and bore equal; and on account of the long stroke may have more mechanical friction per unit of mean effective pressure, but while the cooling surface per heat unit is increased, the difference in temperature during the last third of the working stroke is so slight that an additional transfer of heat seems improbable.

The exhaust of an engine with an expansion stroke 50 per cent longer than its compression stroke shows but a trace of flame, which is evidence of complete combustion in the cylinders, where it should take place. The reduction of the terminal temperature is about 1000 deg. F. If the thermal efficiency is increased, the disposition of the rest of the heat is immaterial, although observation indicates that, while the losses in friction and water-jacket are but slightly increased, the exhaust loss is considerably decreased.

The average heat balance of five full-load tests of a 50-hp., 10 by 20-in. engine, cutting off, that is, closing the inlet valve, at about two-thirds stroke and using the high value of fuel, is as follows:

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Useful work (thermal efficiency)	26.02
Engine friction	4.86
Radiation and exhaust	22.59
Water-jacket	46.48

This high efficiency would be obtained only at full load or with an open throttle, a condition that seldom occurs in an automobile engine.

EFFICIENCY AT PARTIAL LOAD

The ordinary engine is more efficient at full than at partial load. Unfortunately the automobile engine operates at partial load most of the time; therefore if we would increase the brake horsepower per unit of fuel or the miles per gallon, it should be so designed that its thermal efficiency is greatest with average or partial loads, or at least is no less than with full load.

Fig. 4 shows the diagram from the ordinary automobile engine developing part load with the ignition advanced to compensate for the slow burning mixture; *AB* represents the atmospheric line and the piston travel; *BF* the rarefaction during the induction stroke; *FC* the compression, *CD* the firing, and *DA* the expansion line. The shaded portion shows loss due to back pressure in every cylinder when the engine runs at part load.

As the area of the shaded part of the diagram, or the negative load, must be offset by a like amount of area above the atmospheric line, the fuel consumption of an idling engine is necessarily great. At average load the automobile engine in which the intake is throttled has some back pressure, so that the thermal efficiency must be low.

The loss due to low compression and caused by the greater ratio of surface total heat units liberated manifests itself when a heat balance is obtained.

Tests have demonstrated that an engine driving a car on a smooth road at 20 m.p.h. will require no more fuel than when idling at the same speed. The pressure in the inlet manifold when the car is operating at about one-half its maximum speed, say 20 m.p.h., averages about 8 lb. absolute; therefore about 7 lb. more mean effective pressure than the net load requires is necessary under the average condition. When idling, an engine can be working against from 10 to 12 lb. back pressure, a sufficient cause for its high fuel consumption.

Tests of a 4½ by 5½-in. six-cylinder engine made in 1912 by the Automobile Club of America* showed the following thermal efficiencies: Full throttle (1113 r.p.m.), 17.8; one-third throttle (1039 r.p.m.), 15.1, and one-sixth throttle (266 r.p.m.), 6.2 per cent.

IGNITION GOVERNED BY SPEED AND LOAD

As the load becomes lighter, the compression pressure less, and the mixture weaker because of the larger percentage of carbon dioxide present, the inflammation is necessarily slower, and earlier ignition is necessary to obtain maximum efficiency under adverse conditions.

*A Comprehensive Motor Test. Herbert Chase. S. A. E. TRANSACTIONS. Part II, 1912, p. 141.

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The car driver could, if he would, keep his ignition as early as would be consistent with smooth running. In doing so he would maintain the highest possible efficiency. The solution of the problem, in the author's opinion, is to advance automatically the time of ignition directly as the speed and inversely as the load—a combination that will insure the best economy during all ranges of load. With a late spark and slow burning mixture it is possible to generate more heat in the exhaust manifold than in the engine cylinders.

VARIABLE CUT-OFF GIVES HIGHEST THERMAL EFFICIENCY

An advanced ignition and a throttled mixture with resulting low compression will cause the terminal pressure to approach atmospheric, but at partial load the losses from wire-drawing and the low compression are probably more serious than the exhaust-gas losses at full load.

The wire-drawing loss shown in Fig 4 can be prevented by closing the admission valves earlier instead of throttling the mixture. This

FIG 4—PARTIAL-LOAD DIAGRAM

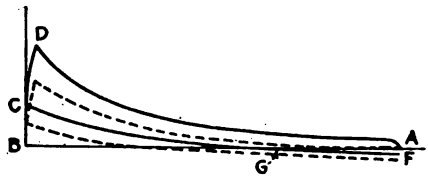

FIG. 5—DIAGRAMS WITH TWO-THIRDS CUT-OFF

will reduce the mean effective pressure without creating a back pressure and will eliminate the light load losses, except those inherent to low compression.

If the maximum cut-off is two-thirds the working stroke (Fig. 3), other conditions being the same, and we make this cut-off earlier as the load decreases, we will maintain a higher thermal efficiency during the whole range of load than is possible in the automobile engine controlled by a throttle in the suction pipe.

WEAKER MIXTURE AS COMPRESSION INCREASES

A carbureter properly designed for efficiency should provide a weaker mixture as the compression increases. A mixture too weak to ignite at atmospheric pressure will fire and do work if sufficiently compressed. A rich mixture is necessary in starting and with partial load, because the exhaust-product content being constant and the compression less, the combustible molecules are not sufficiently close for proper chemical action, therefore more of them should be in the same space to get rapid and efficient firing.

Blast furnace gas having as low as 90 B. t. u. per cu. ft., making 45 B.t.u. when mixed with a like amount of air, will not burn at atmospheric pressure, yet makes an ideal fuel when compressed to 180 lb. gage. Air containing less than 50 B. t. u. per cu. ft. of gaso-

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line vapor will not burn at atmospheric pressure, but when compressed sufficiently will ignite and do useful work.

If the molecules of a hydrocarbon are driven by compression too close to the molecules of oxygen, the generated heat will cause spontaneous combustion, but with a weak mixture high compression is necessary to bring them to a normal ignitible relation; therefore compression can be increased inversely with the heat units supplied. In a Diesel engine, in which the heat of compression is always sufficient for ignition, combustion takes place even with the smallest possible injection of fuel.

It is evident, therefore, for efficiency in automobile engines that the fuel per unit of air should be less as compression increases to prevent spontaneous combustion; conversely for a high mean effective pressure it should become richer as the pressure decreases.

ADVANTAGES OF HIGH COMPRESSION

The higher the compression, other things being equal, the greater the thermal efficiency; yet the compression must be below the critical

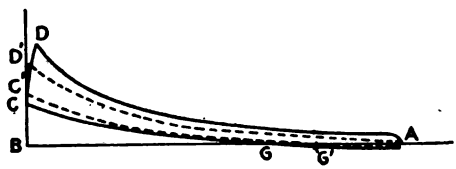

FIG. 6—EFFECT OF VARYING CUT-OFF POINT

ignition point. If we weaken the mixture as the compression increases, the relation of the combustible molecules will remain constant, maintaining practically the same rapidity of inflammation.

If an engine is designed to give maximum compression and rated load at two-thirds cut-off, as shown in full-line diagram, Fig. 5, a better efficiency will be obtained at full load than is obtained in the ordinary automobile engine, because less heat goes out with the exhaust and more is turned into work.

It has already been shown that if point *G* is moved towards *B* as the load gets lighter, there will be no loss from wire-drawing, but as the compression decreases the efficiency will become less, the molecules further apart and inflammation slower. Therefore the ignition must be advanced to obtain the dotted diagram, in which the firing line is practically parallel to *CD*. Such a method of governing would materially increase the efficiency at half or partial load, which is the load most usually required of an automobile engine.

An internal-combustion engine with no compression and firing at atmospheric pressure requires about five times as much fuel per brake-horsepower hour as an engine compressing to five atmospheres and running on the Otto cycle, so that the desirability of high compression can readily be seen.

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If, instead of moving the point of cut-off *G* towards *B*, leaving the mixture practically constant but taking in less gas as less load is required, we move the cut-off point *G* towards *A*, or to *G'*, Fig. 6, and at the same time decrease the amount of fuel per unit of air, the compression will be increased, inflammation will occur just as rapidly, premature ignition will be improbable, the thermal efficiency should be practically as high as at full load and the intial pressure should be less, insuring smooth running at light loads.

FUEL ADJUSTMENT REQUIRED

If the amount of fuel is diminished as the compression increases so that there will be just enough fuel at maximum compression to

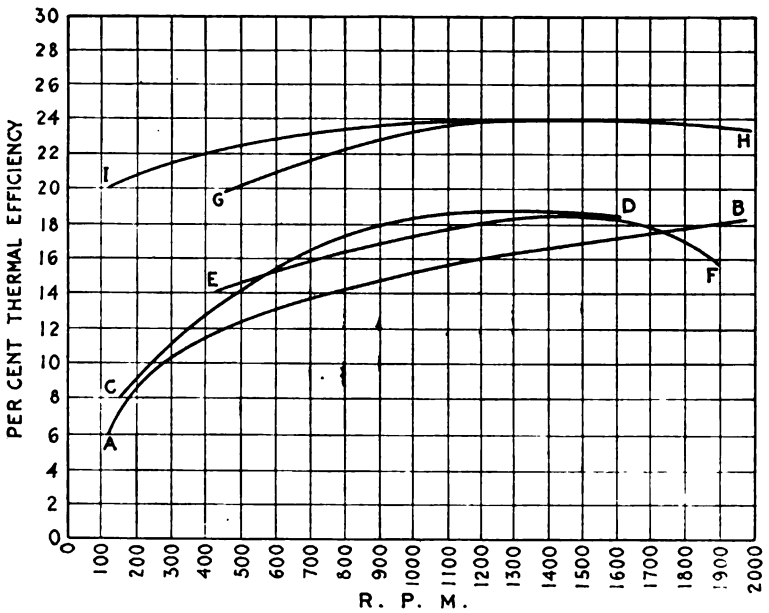

FIG. 7—THERMAL EFFICIENCY CURVES OF INTERNAL-COMBUSTION ENGINES

drive the engine idle, and at the earliest cut-off to give the maximum mean effective pressure, the adjustment is complete. Should the fuel diminish faster than the compression increases at the latest cut-off, there would be a negative mean effective pressure of compression and the engine would stop. Between extremes, full and no load, the fuel compensation should be automatic, thus securing complete and smokeless combustion.

Such a cycle provides for a maximum or at least an average efficiency through the whole range of load. It eliminates losses resulting from wire-drawing and low compression. It provides a fixed fuel ad-

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justment for full load with only speed compensation required. As the load becomes less the fuel valve is throttled and the closing time of the inlet valve retarded, a most simple mechanical problem. The molecular arrangement is such that the inflammation occurs with practically the same rapidity at all loads. The time of ignition, therefore, need be advanced only with the engine speed.

With such a cycle, the theoretical compression depends upon the volumetric efficiency (ratio of volume corresponding to point where the compression and atmospheric lines intersect to stroke volume). This efficiency can be maintained at unity, as the volume of induction stroke can be varied readily.

If the compression at two-thirds cut-off and maximum load is 90 lb. absolute, the maximum compression at no load would be practically 144 lb., yet so attenuated would be the mixture that premature ignition would be impossible.

The thermal efficiency curves *AB*, *CD* and *EF*, Fig. 7, were obtained from tests made of automobile engines by the Automobile Club of America, and probably show the average thermal efficiency of the majority of such engines.

The efficiency curve that would probably be obtained from an automobile engine in which the expansion is 50 per cent longer than the induction stroke, but in which cut-off takes place earlier as the load gets lighter, is shown at *GH*, Fig. 7; *IH* is the theoretical efficiency curve of a complete-expansion engine, in which the compression is increased inversely, and the fuel supply directly, as the load.

SOLUTION OF MECHANICAL DIFFICULTIES

In solving the mechanical problem involved in building an engine to run on the cycle outlined provision is made for retarding the beginning of compression when cranking, thereby introducing no starting difficulties with the normal high compression.

A standard carbureter would require no change in adjustment for full load, but would require a fuel opening varying inversely with the amount of air induced. The fuel valve and cut-off lever would be connected together and operated by the accelerator pedal or hand lever on the steering wheel.

The rarefaction of the mixture in the cylinder at full load induces expansion, gasification and a thorough commingling of the combustible molecules before compression, a condition insuring rapid ignition and complete combustion.

An engine smokes usually because pressure in the crankcase drives the oil past the piston to the combustion chamber during the rarefaction developed with a closed throttle and light loads. As more oil is required at full load than when the engine is idling, the cycle proposed promises more rarefaction at full load than at light loads, insuring a proper oil distribution during the whole range.

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The thermal efficiency of complete-expansion engines has been excelled only by engines of the Diesel type. Unforeseen difficulties may arise in increasing the compression as the load decreases, yet the possibility of developing an automobile engine whose thermal efficiency is at least from 3 to 5 per cent higher than that of the best engines known, and one in which the efficiency is as high at the partial load at which it is usually run as at full load, should be of vital interest to the automobile industry.

DISCUSSION

C. P. GRIMES:—One difficulty with a variable cut-off is that it requires a multiple-inlet manifold. I have observed from experiments that a multiple-inlet manifold will not operate satisfactorily. The same area of passages must be used at both low and high engine speeds. When idling with relatively large passage area and low velocity of charge, the throttling suction, 18 in. Hg., is needed to reduce the boiling point of the fuel to the temperature at which the greater portion of it will remain a true vapor and be carried to the cylinder. I have found that in a horizontal plain-tube carbureter operated under low manifold suction, 1½ in. Hg., up to one-half its capacity, over 60 per cent of the fuel drawn from the nozzles will not be vaporized or carried along by the air-stream but will drop out and accumulate on the bottom of the passages. The same instrument seemed to work very well on a car under throttling conditions.

C. E. SARGENT:—Carburetion is assisted by rarefaction, but if we were controlling engines with a cut-off we could provide other means of gasification. The rarefied gas happens to be present in the inlet manifold, and that is why we depend on it.

DIESEL CYCLE FOR AUTOMOBILE ENGINES

HOWARD C. MARMON:—What are the possibilities of using a Diesel cycle on these smaller engines?

C. E. SARGENT:—Some years ago I thought the only way to start an automobile engine was with compressed air. Many air starters were tried, but difficulties were numerous. The necessity for holding compressed air at but 200 lb. pressure was one of the reasons air starters were abandoned. The Diesel engine is effective only when operating with compressed air up to 1000 lb.

I do not believe we will be able to maintain that pressure in a vibrating car, and without it successful results from a Diesel engine are doubtful. At full load, a stationary engine using the cycle proposed in my paper showed a thermal efficiency within 2 or 3 per cent of that of the Diesel engine, thus maintaining approximate Diesel efficiency in automobile engines without carrying air at 1000 to 1200 lb. pressure.

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HOWARD C. MARMON:—Is not the arrangement suggested in the paper obtained more by mechanical control of the valve timing than by the regulation of the fuel supply?

C. E. SARGENT:—If the full-load cut-off is two-thirds of the stroke, then the cut-off is advanced as the load gets lighter, and the fuel is reduced by the same operation.

VALVE CUT-OFF AFFECTS CARBURETION AND FLEXIBILITY

A. P. BRUSH:—The first four-cylinder engines of a well-known manufacturer had cut-off valves, but even though fuel grades were higher, the low vacuum in the intake manifold made it difficult to reach the evaporation point of the fuel. The intake flow had a fair velocity, but was too intermittent and the handling of the fuel was not satisfactory.

It is not unusual to close the inlet valve 50 to 60 deg. past lower dead center, so that when the engine is turned slowly, compression takes place only as shown in Mr. Sargent's illustrations. But the late inlet-valve closure is due to the development of the high-speed engine, and is one of the difficulties I foresee in carrying out Mr. Sargent's cycle. The usual lag of the inlet-valve closing would also be obtained if the cut-off were 50 deg. before the lower dead-center, as suggested. But the time for getting the gases into the cylinder is thereby 80 per cent less than with the conventional practice. This would seriously handicap the flexibility of the modern engine.

Considerable work has been done relating to the spark advance Mr. Sargent advocates. The spark advance due to speed does not answer all the requirements in the automobile engine, because compression is also a factor of spark advance. One of the most ingenious methods of obtaining spark advance is by varying the timing in proportion to the vacuum in the manifold.

INTERMITTENT FLOW OF MIXTURE

C. E. SARGENT:—When the cut-off is earlier as the load gets lighter, there is a long space between charges, but if we cut off at two-thirds stroke at full load, and later, as the load gets lighter, the manifold could probably be so designed that the inertia of the gas as it is cut off would increase its pressure for the next cylinder. A cut-off control would not be as undesirable as it was in the earlier engines. The engine speed would be harder to control, but the cut-off could be adjusted by hand at the steering wheel.

Cut-off engines were made some years ago, but it was much cheaper to make a butterfly valve than a cut-off mechanism. I think that was one of the reasons they were abandoned.

R. H. COMBS:—In 1909, six-cylinder high-speed marine engines using the four-stroke cycle were built in New York, so thoroughly under control that they were reversible, the valves being controlled by lugs. The throttle, valve-gear control, and compressed-air valves for starting were all on the same lever.

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HOWARD C. MARMON:—The cut-off valve engines were not properly adjusted because of the high vacuum necessary to reach the boiling point of the fuel. That is not a valid objection, because so much heat is wasted that could be used to raise the temperature and thus to reach the boiling point.

MORE PUMPING RESISTANCE WITH HIGH COMPRESSION

C. P. GRIMES:—An engine idling, or working at slow speed, must generate enough power to overcome the friction resistance plus the pumping resistance, because it is always sucking in air, compressing it, and discharging it as burned gas. If the intake pressure is restricted to 18 in. Hg. before the charge enters the cylinder, the power required to turn the engine will not be as much as if the charge were unrestricted while entering, and were compressed to a higher pressure. I doubt if the added efficiency would much more than overcome the additional resistance due to pumping, even were it possible to carburet the proposed cycle.

Since we are attempting to gain greater efficiency at low and ordinary operating speeds, I would like to relate the method adopted by Geo. Schebler for improving the power factor of his twelve-cylinder engine. At ordinary speeds the throttle of only one block is used. When in a hurry, he uses both, thus working the two six-cylinder blocks at a good power factor.

C. E. SARGENT:—With the cycle proposed, the compression would be high while idling, but of course the compression is always given back, so that, while the angular velocity variation might be unusual, there is no question as to the increased efficiency of the cycle.

JOHN O. HEINZE:—If we could reduce the compression space as the power of the engine was reduced, high compression would be maintained, regardless of speed and power, and therefore maximum efficiency of the fuel would be developed. Apparently it is not possible to accomplish this at the present time.

THERMAL EFFICIENCY AND HORSEPOWER

C. E. SARGENT:—We have been closing the inlet valves late in order to get maximum power from the engine at a sacrifice of thermal efficiency. Late closing at high speed permits the gas to flow into the cylinder after the piston reaches its center, giving maximum capacity, which we will have to forego in order to gain efficiency. The great desideratum is to get efficiency at light loads, at which the engine is usually run, and, if we must have capacity, increase the stroke and the weight of the engine.

JOHN O. HEINZE:—Some years ago we built an engine with sliding cams that controlled the closing and opening of the exhaust and inlet valves, in this way varying the cut-off. When we operated this engine at 1000 r.p.m., and manipulated the cam, we obtained a thermal efficiency of 19 per cent. The horsepower developed at this efficiency was not the maximum horsepower the engine would develop at lower thermal efficiencies. The horsepower output depends on the volume of gas consumed and on its efficient mixture.

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INLET VALVES WITH AIR CYLINDERS

CHAS. TRASK:—About ten or twelve years ago, a Winton engine had air cylinders above the inlet valves.

CHESTER RICKER:—That engine had an automatic valve with a light spring, only strong enough to lift it and hold it on its seat. Attached to the valve stem was a piston. Beneath the piston air pressure could be introduced, and that would in effect increase the spring tension on the inlet valves. The engine was throttled with a small outlet valve, which, when closed, gave maximum pressure, and held the valves tightly on their seats, so that the atmospheric pressure barely lifted them. To open them, the pressure underneath was reduced until maximum speed was reached, when the springs alone were active.

C. E. SARGENT:—What is needed is a quick Corliss-type cut-off, especially when the piston is traveling at high speed.

PROPOSED CYCLE GIVES RESTRICTED PERFORMANCE

A. P. BRUSH:—The modern automobile engine has been developed from the standpoint of the pleasure of the user, and the essential question seems to me to be whether a gain in efficiency would offset the distaste of the public for the performance so restricted that would result if the time the inlet-valve remains open be decreased 40 per cent.

C. E. SARGENT:—I cannot see how the performance would be less satisfactory. At full load we have three-quarter cut-off. Now, we reduce the mean effective pressure by throttling the mixture in the intake pipe. It is only a question of reducing the mean effective pressure, in this case, by letting in less fuel. I do not see why the mean effective pressure cannot be reduced as well with this cycle as with that used in the ordinary engine.

JOHN O. HEINZE:—You would hardly expect to get as much power with 5 cu. in. as with 10 cu. in. piston displacement, even though the mean effective pressure was less with the latter displacement.

C. E. SARGENT:—With the same piston displacement, the power would be less, but the efficiency would be greater. The necessary power could be obtained with a longer stroke.

EFFECT OF VACUUM ON SPARK DISCHARGE

C. P. GRIMES:—The effect of a vacuum on a spark discharge is shown by the fact that in the Moore vacuum lighting system a spark is often discharged through 20 or 30 ft. of glass tubing, while at atmospheric pressure the same spark would not pass a gap of more than a few inches. It is often difficult to pass a spark at slow engine speeds, even when the discharge takes place in a low-pressure medium. I believe the difficulty would be greatly increased if a compression of 60 or 80 lb. were used, as proposed by Mr. Sargent.

JOHN O. HEINZE:—A spark that will penetrate compressions of 300 to 400 lb. is easily produced. It is simply a question of cost of the apparatus.

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ELECTRIC TRANSMISSION ON OWEN CARS

By J. B. Entz

(Member of the Society)

Abstract

This paper contains a brief description of the Entz electric transmission. Wiring connections are given of the several speeds, for electric braking, for starting the engine and for charging the battery. The statement is made that the electric transmission eliminates and does the work of the friction clutch, the clutch pedal, the transmission gears, the flywheel and separate starting and lighting systems.

The electric transmission as installed in Owen cars consists of two direct-current dynamos and a drum controller. One of the dynamos has its field-magnet frame directly connected to the engine crankshaft, the frame taking the place of the ordinary fly-wheel. This machine acts as both a clutch and a generator; its armature is mounted on a large hollow shaft. The second dynamo has its armature mounted on the same hollow shaft as the first; its field-magnets are stationary and it is generally used as a motor to help drive the propeller-shaft, and to boost the effort of the engine as transmitted through the generator, which like any clutch, can only transmit the engine effort or torque. The assembly, Fig. 1, shows the arrangement of parts; collector-rings are used to convey the current from the generator as all of its parts revolve.

WIRING CONNECTIONS FOR VARIOUS SPEEDS

In this transmission the control is accomplished by one lever on top of the steering-wheel, which simply moves a drum controller to the different positions, between which there can be no interference.

Figs. 2 to 8 illustrate the connections for the various positions of the drum controller. The gradations of speed and torque are controlled by the relative strength of the generator and the motor fields. The weaker the generator field compared to the motor field, the greater the slip between engine and propeller shafts and the more electrical energy goes to the motor, thus producing greater torque.

In Fig. 2, showing the first control position, the generator field is shunted and weakened while the motor field is at full strength. Fig. 3 shows both fields unshunted, but the motor field is still the stronger because it is wound with more turns. Fig. 4 shows the third position with the generator field unshunted. The motor field is shunted so that it is weakened compared with the generator field. It then gives less torque for a given current, but also less counter-electromotive force and therefore less slip at the generator.

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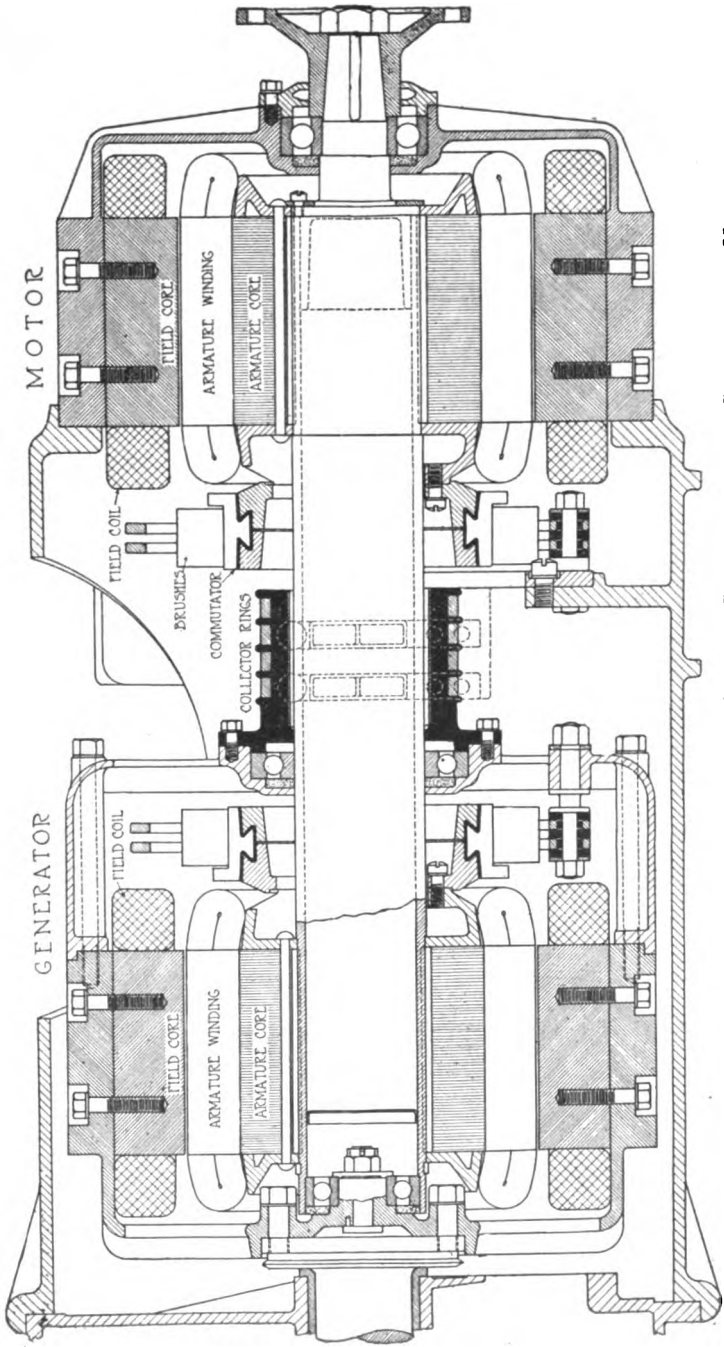

FIG. 1—ASSEMBLY OF ELECTRIC TRANSMISSION SHOWING SECTIONAL VIEW OF GENERATOR AND MOTOR

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In the fourth and fifth positions the motor field is gradually weakened by regulation of the controller, but the external connections are the same as shown in Fig. 4. Each of these steps approaches more closely to the high-speed position, Fig. 5, where both the armature and the field of the motor are cut out of the generator circuit. The generator is used as a clutch only in the high-speed position,

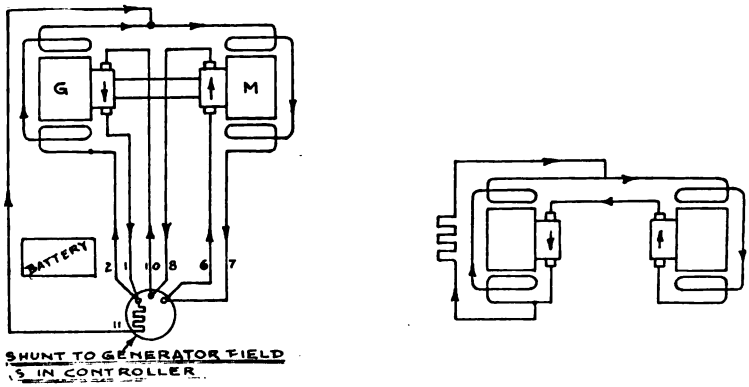

FIG. 2—WIRING CONNECTIONS FOR FIRST SPEED

when its armature is short-circuited; a small speed-difference (or slip) between the armature and the field is necessary to establish the current in the generator windings that energizes and causes the generator to act as a clutch.

The motor in this position has a shunt field, opposed by a series field in the battery circuit, thus giving it a differential-field connec-

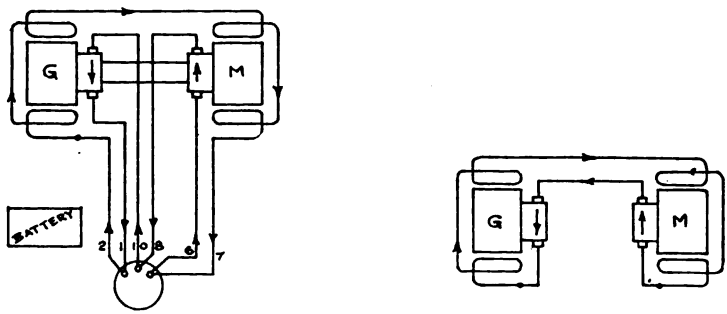

FIG. 3—WIRING CONNECTIONS FOR SECOND SPEED

tion with an inherent self-regulating characteristic. On power-control positions other than the high-speed position, the motor helps to turn the propeller-shaft by taking current from the generator; the slip in the generator is then greater than needed to energize the clutch and the current thus produced is used in the motor to give

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additional turning effort to the propeller-shaft. In the high-speed position the motor plays no part in the transmission of power but acts as a charging generator for the storage battery, which supplies current for the engine starter and the electric lights.

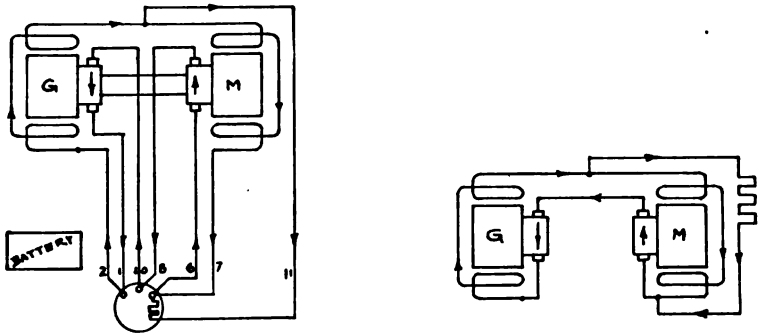

FIG. 4—WIRING CONNECTIONS FOR THIRD SPEED

Besides the power-control positions, there is a neutral position, Fig. 6, in which the clutching effect is cut out, but in which the motor is connected through a resistance so as to act as an electric brake. The motor then becomes a generator, taking power to drive it, thus

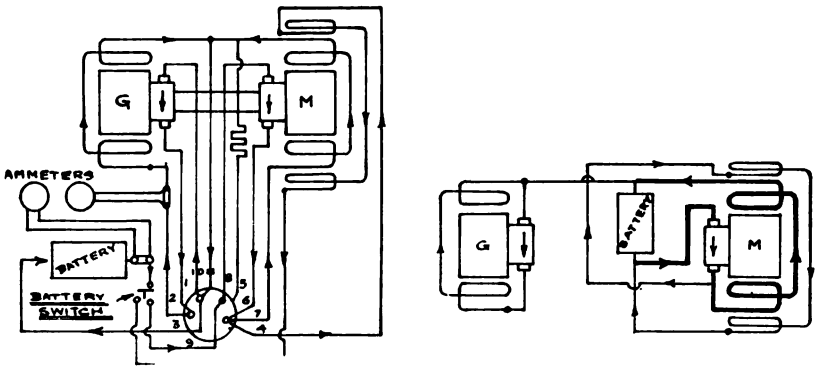

FIG. 5—WIRING CONNECTIONS FOR SIXTH SPEED

retarding the car. This brake will hold the car on any mountain grade at a speed of 20 m.p.h., without wear of parts, and can be applied with the car going 60 m.p.h. It cannot hold the wheels, and there is little danger of skidding, as the braking effort disappears at speeds below 15 m.p.h.

Fig. 7 shows the connections when the generator is used as an engine starter; the field and armature connections are reversed so as to turn the engine forward, and the starting battery is included in the generator circuit.

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Fig. 8 shows a way of charging the battery while the car is standing; current can be supplied from the generator at a much higher rate of charge than would be safe to establish for running conditions. If the battery should become run down, it can be brought

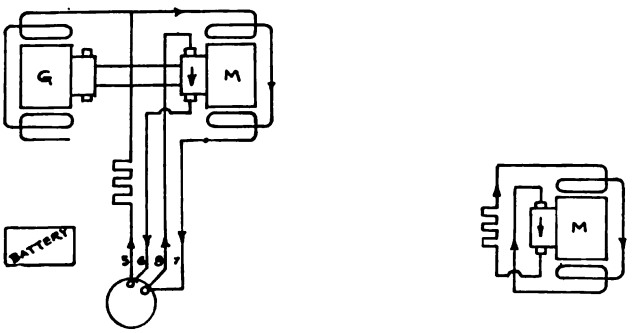

FIG. 6—WIRING CONNECTIONS FOR ELECTRIC BRAKING

up in a short time, as a high rate of charge is permissible for a battery that has not reached the gassing point and is not warm. The engine speed is so low that there is no danger of the water

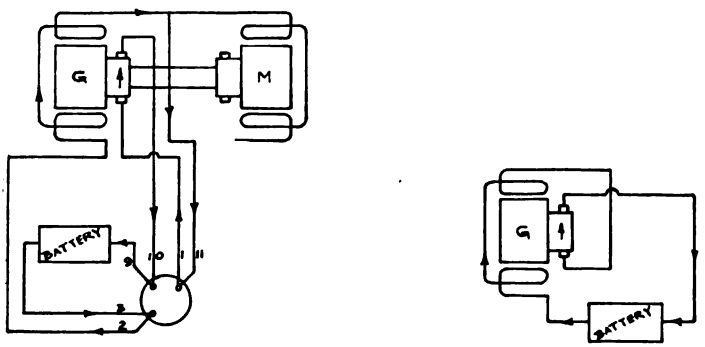

FIG. 7—WIRING CONNECTIONS FOR STARTING ENGINE

boiling in the radiator. It is a way of giving a boosting charge that is at times very desirable.

OPERATING ADVANTAGES OF SYSTEM

The turning effort impressed on the propeller-shaft is as uniform

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and smooth as that from an electric motor; in fact, it is just the same. No shocks or jars can be transmitted from the engine, as there is no mechanical connection between the engine and the driving shaft.

From the time of starting the car until maximum speed is reached, and through all the range of power required on level road or the worst hills, the power between the engine and the propeller-shaft is never interrupted. The small lever on top of the steering-wheel controls the car in traffic, and on winding irregular grades calls forth all the power of the engine at just the instant and as long as it is needed. The car can be held on a grade by its engine power, the generator slipping and holding with the aid of the electric motor, ready at once to go forward upon opening the throttle; by closing the throttle slightly the car can be allowed to go backward, then held, and then permitted to go forward again at the maximum speed the grade permits.

From standing with the engine idling the car can be brought smoothly and rapidly up to 20 to 25 m.p.h., the speed of city traffic. Acceleration can be accomplished without any speeding-up of the engine previous to dropping in the clutch and without gear-changes made at high engine-speed, as in the case of a rapid getaway with a gear-transmission car.

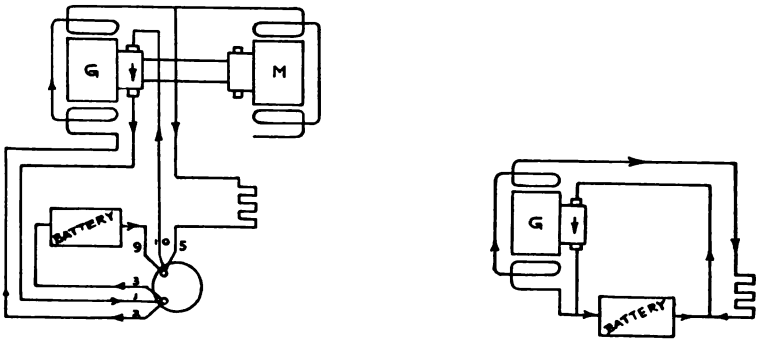

FIG. 8—WIRING CONNECTIONS FOR CHARGING BATTERY

The electric transmission eliminates and does the work of the friction clutch; the clutch pedal; the transmission gears; the fly-wheel; separate starting and lighting systems; and the clutches, gears, chains, connections and automatic devices.

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DISCUSSION

PROF. D. L. GALLUP:—How much of the engine power is delivered by the Entz transmission at the rear wheels, the system working at its highest efficiency for the various speeds? Naturally everybody is interested in flexibility, but not to the extent of sacrificing a good proportion of the gasoline in the form of power.

I would also like to inquire about the "surging" effect when traveling at slow speeds.

J. B. ENTZ:—The loss in the windings is about 5 per cent. The friction in the brushes, etc., helps transmit a small part of the engine torque mechanically. On an ordinary level road, the torque required calls for a slippage of 50 to 55 r.p.m. The maximum engine torque requires a slippage of 92 r.p.m., as measured. This slippage is the working loss. As long as the engine runs at a speed below that corresponding to the peak of its torque curve, more power is delivered to the propeller-shaft on account of the slippage. At the maximum available speed, as the engine torque falls off, the slippage of 92 falls off to somewhere around 75 or 80 r.p.m. The efficiency at 1000 r.p.m. will be 92 per cent, the 8 per cent representing a speed loss, with no friction loss in the gear box.

NO SURGING UNDER LOAD

There is absolutely no surging when pulling a load. At the high-speed position, the car can be run down to very low speed, because there is a flexible connection between the engine and axle. If you are driving on the level and put the lever in a lower-speed position of the controller, the load on the engine is thrown off, and the car coasts until its speed is lower as compared to the engine speed than before, when the load is picked up smoothly and without intermittent driving effort.

R. W. KNOWLES:—Is the reversing done mechanically or electrically? How much more does the electric transmission weigh than the clutch, starter, transmission, etc., in a comparable car? How much more is the cost of an electric transmission than that of a similar grade of mechanical transmission?

What happens if the controller is closed with the engine at fairly high speed and the car at a standstill? Since more current must be delivered to the motor to get acceleration, the voltage must be higher. Does this not mean greater slip?

J. B. ENTZ:—Experience has taught us that the reversing can be done better mechanically. There is no objection to the simple shifting of a lever while the car is at a standstill.

We have weighed the two types of transmission and found no difference. We dispense with the flywheel, the separate starting and lighting systems, the clutch and the transmission gears. Many accessories not ordinarily weighed are eliminated.

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The price depends on quantity of production. I cannot give the exact cost because I do not know the shop-production figures. We can assemble the engine and all the separate parts in a car in 2½ hours.

More current is taken while accelerating. A certain current in the generator represents a certain torque. When more torque is required on the propeller-shaft than the engine itself has, the motor must be included in the circuit, and the generator acts only as a clutch. It can only transmit the engine torque. I have given the slip in the generator for maximum torque, which is doubled on the lower speeds when the motor is included.

If the engine should be running at 800 or 1000 r.p.m., and you should suddenly close the controlling lever, no damage whatever would result. The electrical system is entirely protected, because the maximum torque can only produce a certain current, which the apparatus can transmit continuously.

EFFECT OF OPENING THROTTLE WHEN COASTING

CHAIRMAN F. E. MOSKOVICS:—When I was running a car with the electric transmission, about the most dangerous thing I could do was to put the lever in neutral at 50 m.p.h. I nearly went through the windshield. When the car was coasting, and I opened the throttle suddenly, I could feel a bang.

J. B. ENTZ:—The clutch action cannot take place until the generator, which acts as a flywheel, is running faster than the propeller-shaft. If you take your foot off the throttle and the engine drops in at a speed of 100 r.p.m., you run freely. Now if you open up again, the engine itself accelerates freely, except for the effort necessary to accelerate the flywheel. It does not build up until the engine speed exceeds that of the propeller-shaft.

The car has to be driven in order to appreciate its advantages. The first time one drives it, there is a tendency to speed up the engine and drop in the controller, in some such way as with a clutch. It is best to put the controller in the first position with the engine turning slowly (it will not stall as slippage occurs) and then open the throttle and get away.

W. G. WALL:—The losses in this transmission can be figured easily. It would be interesting, however, to drive an electric-transmission car on the Speedway and measure its acceleration and speed in order to compare it with an identical car with the ordinary transmission. Including the efficiency, and the difference in weight, some direct figures would be had.

The slippage that prevents injury to the car when closing the circuit is an advantage most of the time. It is advantageous also, when in a tight place, to be able to throw in the clutch quickly and to

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have the momentum of the flywheel carry the car out. It seems impossible to do that with the electric arrangement—the effect would not be the same. The same amount of energy would be used, but the results would be different.

J. B. ENTZ:—We can get a car out of a hole by a method equivalent to speeding up the engine, but much more efficiently. When a generator furnishing energy, most of which is utilized through an outside circuit, is short-circuited, more effort will be put on the engine, as well as on the shaft, than is exerted by any form of mechanical clutch. With the friction clutch one can speed up the engine to 500 or 600 r.p.m., let in the clutch, and when the engine speed drops down to 200 or 300 r.p.m., it stalls. With the electric transmission, dropping into the same speed, the maximum drawbar pull is maintained with the car stationary and the engine turning.

SOME OPERATING EXPERIENCES

G. A. WEIDELY:—In the latter part of 1914 I drove to Indianapolis from New York, using a car with the conventional gear transmission. R. M. Owen drove a car similar to mine, except that it was equipped with the Entz transmission; except for the transmission unit, the two cars were as near alike as two cars could be; both were equipped with wire wheels, cord tires and had the same type of body. When the top of one was up, the top of the other was also, and when the glass was put down on one, the same thing was done on the other. We alternated, sometimes two or three times a day, in driving the cars. We traveled about 1059 miles. The running time was slightly under 39 hours. Except during the afternoon of the last day, it rained all the way. If I remember correctly, the car I drove was 42 lb. lighter than Mr. Owen's. That may have been due to the transmission.

On level road there was little difference in the speed of the two cars. Sometimes I passed Mr. Owen and sometimes he passed me. Both cars could make 63 or 64 m.p.h. At times I found a hill on which I could hold my own because it happened to be just right for my gear ratio. Mr. Owen beat me on most of the hills, because he could find a ratio that just suited them. There was so little difference in the performance otherwise, including the amount of gasoline consumed, that we could have mixed our figures and not have told the difference. It is likely that the car with the electric transmission used a great deal more gasoline than I did going up hill, but it used much less coming down.

FLEXIBILITY OBTAINED BY LIGHT WEIGHT

CHAIRMAN MOSKOVICS:—The last two years have brought great development in lighter construction, through lessening the number of parts, combining them, and through the more extensive use of alu-

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minum and its alloys. Few high-grade cars use roller clutches, even in electric lighting and starting systems. The use of chain to drive lighting and starting devices is practically discontinued; the generator is simply coupled to the magneto driving-shaft. The result is that the net weight is very light. The electric is simply a one-speed and reverse gear transmission. I believe the question comes to this: Shall we attain flexibility by extremely light clutches, transmissions, etc.; or shall we disregard weight, relatively speaking, and use the Entz method to get flexibility and the added advantage, which I think we will all concede, of smooth operation in starting?

J. B. ENTZ:—The lightness of modern cars, except as attained by extensive use of aluminum, can be had in one car as well as in another. The light weight is due to the use of the high-speed engine with timing and valves such that it will hold its torque at very high speeds. For a given horsepower the torque is, say 100 lb. in. at 2000 r.p.m., whereas we formerly had 200 lb. in. torque at 1000 r.p.m. The reductions to-day are 4½ or 5 to 1, whereas they used to be 3 to 1. This electric transmission will make the best showing with the high-speed engine. It is not a horsepower but a torque transmission.

D. S. HATCH:—Are there any advantages in making the clutch a combination friction and electric clutch, instead of completely electric?

J. B. ENTZ:—I think that the combination clutch was proposed by me, years before any one else made it. I worked with a centrifugal clutch so that there would be no slippage. I found that the cushioning effect is so desirable that the straight electric-transmission is much better. Electric-transmission cars built in 1906 and 1907 have run 200,000 miles, have never had any oil put in the transmission, never had a brush repaired, and are running to-day.

FUTURE CAR OF HIGH POWER AND LIGHT WEIGHT

G. P. DORRIS:—The future car will be of greater power and less weight. At present we have engines of excessive power in order to satisfy the demand for cars that will operate on high gear practically all the time. That leads in a way into Mr. Entz' hands. We can, however, go still further in simplifying the present gear transmission; we can reduce its size and weight by using only two speeds, eliminating the gate and the selective mechanism. The flexible feature of the drive can be obtained by introducing a member between the transmission and the rear wheels such that the car would act as though pulled by a rubber string.

Few drivers use the full ability of the present cars, which are practically worn out at moderate speeds and on good roads. As this average condition is likely to continue, a high power light-weight car of two speeds may be a future possibility.

CHAIRMAN MOSKOVICS:—The trend seems somewhat the other

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way. The designers of multicylinder engines discuss the elimination of the gearbox, but I do not see them abandoning it.

F. A. CORNELL:—It is said that the fuel consumption of the electric-transmission cars is excessive in ordinary city driving.

J. B. ENTZ:—We have found that the city operation has been, if anything, more efficient. The losses are not 50 r.p.m. in city driving, as there is considerable coasting. Where frequent starts and stops are made, the greatest amount of energy is required in retarding and accelerating. The less power thrown away in brake application, instead of letting the car drift into low speed, the more power is saved. There has been no criticism of the electric transmission because of fuel consumption, or of the lack of power or hill-climbing ability.

TESTS OF CAR WITH ELECTRIC TRANSMISSION

C. P. GRIMES:—This afternoon (April 28, 1916) the Research Committee† of the Indiana Section conducted tests of an Owen car equipped with the Entz transmission. The tests were run on the Indianapolis Motor Speedway, the car being driven by R. M. Owen. They were made to determine the current required at different speeds and during retardation and acceleration. The results obtained are shown in the accompanying tables. The barometer reading during the tests was 29.39 in. of mercury and the temperature was 64 deg. F. The gear ratio of the car tested was 3½ to 1; its weight empty, 4540 lb., and loaded, 5170 lb. The road wheels were 35-in. diameter. The current consumption at various speeds follows:

Car speed, m.p.h. ...	10	15	20	25	30	40	45	50	55
Current consumption, amperes.....	35	35	45	52.5	55	55	60	70	77.5

CHARLES GUERNSEY:—What is the lowest possible gear ratio that can be used with the electric transmission?

J. B. ENTZ:—Anything you desire. The average is about 3 to 1, as in a geared car. Our car is geared 3½ to 1. With a car geared at a less reduction than this, we can travel anywhere between New York to Chicago without difficulty. In some of these cars we have put a low-speed reduction, which is equivalent to what is known as a two-speed axle. The presence of two speeds does not mean the transmission is part mechanical and part electrical while running the car, but that the axle gear-ratio can be changed at will; otherwise the control is all electrical as described. The reverse is the same as that used in the ordinary sliding gear transmission.

ENTZ TRANSMISSION USED ON TRUCKS

CHARLES GUERSNEY:—In case a greater reduction is desired than is ordinarily used in standard pleasure-car practice, a mechanical

†This committee consisted of Chester Ricker, Max Thoms and C. P. Grimes, chairman.

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reduction is used. Would it be feasible to make a sufficient reduction for heavy truck work without using the mechanical reduction?

J. B. ENTZ:—We have built several trucks and found it unnecessary to use a greater reduction than in touring cars. The trucks must have a greater total gear-reduction but the range between high and low need not be any greater than in a touring car. A truck may start on a hill at first speed. It runs so slowly that the time taken to throw out the clutch and make the gear change is almost sufficient to

RESULTS OF DECELERATION AND ACCELERATION TESTS

RETARDATION*		ACCELERATION	
Speed Range, M.P.H.	Time, Sec.	Speed Range, M.P.H.	Time, Sec.
15– 5	25.6	0–10	4.3
20–10	31.9	10–20	3.8
25–15	33.7	20–30	7.1
30–20	27.9	30–40	11.9

*These tests were made with the controller lever in the neutral position.

bring the car to a standstill; more damage is done by trying to change from first to second on a grade than in any other way. The truck is running slowly because of the large gear-reduction. In a touring car the momentum will carry it forward. We have taken trucks with trailers where the ordinary truck cannot change from first to second. We can close the throttle to a point which reduces the torque so as to hold the truck stationary, close the throttle still more and let it drift back, then open up to stop it, and finally take it forward. Such control is more necessary with a heavy truck than with a touring car.

CHARLES GUERNSEY:—I differ with Mr. Entz in the matter of truck gear ratios. We can take a truck performing satisfactorily on a high gear of say 8 to 1, put the ordinary reduction as used in pleasure cars, say 3 to 1, on it, and drive through holes, such as contractors and other users often get into, and the engine will not have sufficient power to spin the wheels. I believe the average truck ratio is about 40 to 1, on low gear. Can the performance cited be obtained electrically?

J. B. ENTZ:—It can be done easily. It is just a question of the difference in windings of the two dynamos.

CHAIRMAN MOSKOVICS:—In New York I saw a most interesting demonstration of a truck equipped with the electric transmission. A

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6 by 6-in. timber was nailed to the floor. The truck was moved with its rear wheels against the timber, and was slowly lifted over it, the driver operating only the throttle. The time from starting to lifting over was about 30 sec. and then the wheels were dropped back again.

R. M. OWEN:—During the past winter we have taken out the gear transmission and substituted an electric transmission on several trucks. These are now in operation in Philadelphia and New York.

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METROPOLITAN SECTION PAPERS

FIELD OF FOUR-CYLINDER ENGINES*

By Frank E. Watts

(Member of the Society)

Abstract

The author confines his discussion to engines used on pleasure cars, inasmuch as practically all commercial vehicles use the four-cylinder type. The performance expected of their cars by automobile owners is outlined, particularly as regards performance, durability and maintenance cost. Inasmuch as the horsepower required is often determined by the acceleration demanded, the argument in favor of four-cylinder engines is based mainly on a comparison of its acceleration performance with those of engines having a larger number of cylinders. A number of acceleration curves are given for these engines.

The paper next considers smoothness of operation at low, medium and high running speeds, asserting that the decrease in inertia forces due to lighter reciprocating parts has made it possible to increase the speed and thus reduce remarkably the vibration of the four-cylinder engine. In conclusion, the statement is made that the only proper field for the engines of more than four cylinders is in very high price cars of large weight in which the cost of the engine and of operation is not important. In the medium and low price fields, however, the four-cylinder engine is entirely satisfactory.

During the past year numerous excellent papers have been read dealing with the technique of modern types of automobile engines. Torque variations, unbalanced forces, periodic vibrations, etc., have been called to our attention repeatedly in these papers. Most of this matter is not new; it can be found in the admirable works of Dalby, Clerk, Güldner, Locke, Marchis and Sharpe, most of which have been available for ten years past. Furthermore much of the matter presented is not pertinent, because little attention has been given to the effect the forces described have upon the passengers.

In this paper, therefore, I will consider briefly the applications of engines to automobiles, and try to show why the four-cylinder engine is used on the majority of motor cars produced to-day.

As practically all commercial vehicles, both freight and passenger, have four-cylinder engines, I will confine the discussion to pleasure automobiles. To still further simplify the question, I will consider only the present types of four-stroke poppet-valve engines.

The question to be considered then is: What is an automobile engine required to do? In other words: What is its job?

OWNER'S REQUIREMENTS

Experience gained in the building of some fifty thousand cars, which are now running in all parts of the world, has convinced me

*Presented at joint meeting with New York Section of American Society of Mechanical Engineers.

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that the majority of automobile owners expect their cars to perform about as follows:

To run steadily and pleasingly on high gear at speeds from 5 to 50 m.p.h. over smooth roads.

To carry them over the rougher roads as fast as they can ride with any degree of comfort.

To pull through deep sand, mud or snow easily and without overheating.

To climb any ordinary hill where there are traveled roads, on high.

To get away from a standstill about as quickly as their neighbors' cars.

To do all these things as cheaply as possible, and without skilled care and constant attention.

Naturally the engine's part in securing this performance is important, but the average automobile owner is not greatly interested in the construction or performance of his engine except as it affects the performance, durability and maintenance cost of his car. Let us consider these three factors in the order mentioned.

PERFORMANCE DEPENDS ON COMFORT DESIRED

The horsepower required to give the performance just outlined depends of course largely on the loaded weight of the car. This in turn depends upon the passenger capacity and the comfort desired.

We can divide cars carrying from five to seven passengers roughly into three classes: First, cars of toleration, having a shipping weight of less than 1800 lb. Second, cars of comfort, having a shipping weight of from 1800 to 3000 lb. Third, cars of luxury, with a shipping weight of more than 3000 lb. There are, of course, notable exceptions to this classification. All cars of the first class manufactured in quantity have four-cylinder engines and this undoubtedly will continue to be the case.

It is in the second class that the controversy as to the proper type of engine is most active at the present time. Such cars include a considerable range of sizes. As the horsepower required varies with the loaded weight, we find engines in cars of this class varying from a little more than 20 hp. to a maximum of somewhat more than 50 hp. Most of the engines are included in the range between 25 and 40 hp.

However, the horsepower of an engine is not always an indication of its availability for automobile use. Often the maximum torque and the range of this torque are the determining factors. The maximum torque of the engines I have just mentioned will usually run between 1250 and 200 lb. in. The maximum horsepower does not always decide the availability of the engine for use in ordinary pleasure cars because the speed of these cars should be limited to from 55 to 60 m.p.h. and if the maximum horsepower of an engine is obtained at such a high number of revolutions that it will not be usable at this car speed with a practical rear-axle ratio, it is not of any practical value.

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ACCELERATION DETERMINES HORSEPOWER

At the present time the horsepower required is often determined by the acceleration demanded. Within the last year there has been a tendency to increase greatly the ability for acceleration in motor cars. This has been due partly to the advent of eight and twelve-cylinder cars and partly to the increasing congestion of traffic in our large cities and upon our principal highways, or to other traffic conditions that make quick starting desirable. In this connection some curves taken with the Wimperis accelerometer may be of interest.

Fig. 1 shows the acceleration curve of a popular six-cylinder car, 1915 model. The acceleration at 10 m.p.h. is slightly over 2 ft. per sec. per sec. This car was considered satisfactory at the time it was built and was sold in large quantities.

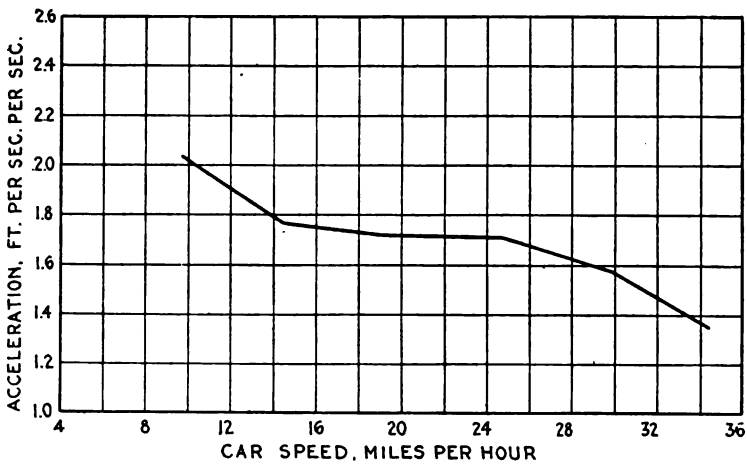

FIG. 1—CURVE OF ACCELERATION WITH 3½ BY 5-IN. SIX-CYLINDER ENGINE

Axle ratio, 49:13. Gross weight, 3500 lb; net weight, 3050 lb. Wind, 12 m.p.h. Top, windshield, curtains up. Road, dry.

Figs. 2, 3 and 4 show accleration curves of well-known and popular makes of four-cylinder cars, also 1915 models. These show accelerations which, while slightly greater than that of the six-cylinder car mentioned (see Fig. 1) are closely comparable.

Fig. 5 shows the acceleration curve of a well-known eight-cylinder European car, which was built especially for city work. The acceleration of this car is noticeably low, but it is the custom abroad to change gears much more frequently than in this country, and this car was fitted with a transmission having four forward speeds.

Fig. 6 shows a comparison of three more recent cars. Cars *A* and *C* have eight-cylinder engines and car *B* has a four-cylinder engine. I believe that the acceleration of car *B* at ordinary driving

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speeds is about as great as is obtained with any stock car fitted with standard gear ratio, at the present time.

Fig. 7 represents a test of the same car as *B*, with the second-speed gear engaged. The rear axle ratio however was 4 to 1, whereas with the standard car this ratio is 4.64 to 1.

COMPARATIVE ACCELERATION ABILITY

A considerable experience in acceleration tests of stock models of American cars purchased in the open market leads to the following conclusions: At the present time the best acceleration, by which I mean the most available for use in traffic conditions, is obtainable from four-cylinder engines. The acceleration of other types of engines at the slower speeds is not usually so quick or as powerful as the acceleration of the four. At speeds of from 15 to 20 m.p.h. and above the performance of different types becomes very nearly equal,

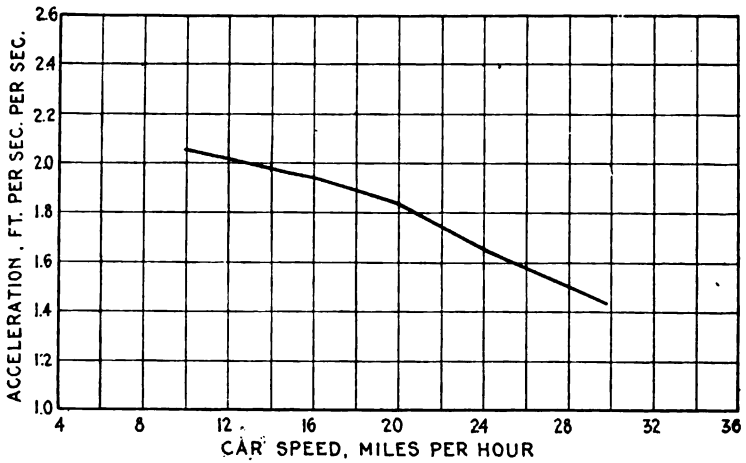

FIG. 2—CURVE OF ACCELERATION WITH 3½ BY 5-IN. FOUR-CYLINDER ENGINE

Gross weight, 2930 lb.; net weight, 2530 lb. Wind, N.W., 20 m.p.h. Top, windshield, curtains up. Road, dry.

and at speeds of more than 30 m.p.h. engines having a large number of cylinders are better than the four. This is particularly true of six, eight and twelve-cylinder engines having single carbureters, which are apt to be rather uncertain at low speed when picking up quickly. The result often is that the more cylinders an engine has the more frequently one is required to use the gearshift lever for city driving. I do not believe that this is a fundamental fault. It is due, I think, to poor carburetion and poor distribution of gas, and will in all probability be remedied.

SMOOTHNESS OF OPERATION

For very rapid acceleration cars having four-cylinder engines are not quite so smooth in action as those in which the power impulses

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are more frequent. But recent changes in four-cylinder design, the decrease in weight of reciprocating parts, the more rigid construction of rotating parts, and above all the very much greater rear-axle gear ratios, which the use of spiral-bevel gears had made practical, have entirely revolutionized the performance of modern four-cylinder engines as compared with those built one or two years ago.

At the present time the difference between a four and a six, eight or twelve can be detected on rapid acceleration, but is unnoticeable to the average driver, unless he is especially looking for it, at any rate of acceleration he will care to pay for. I believe that the desirable limit in acceleration has about been reached, for we have already reached a point where the wear upon tires, and the consumption of gasoline is excessive; the public will not long continue to pay for such high activity.

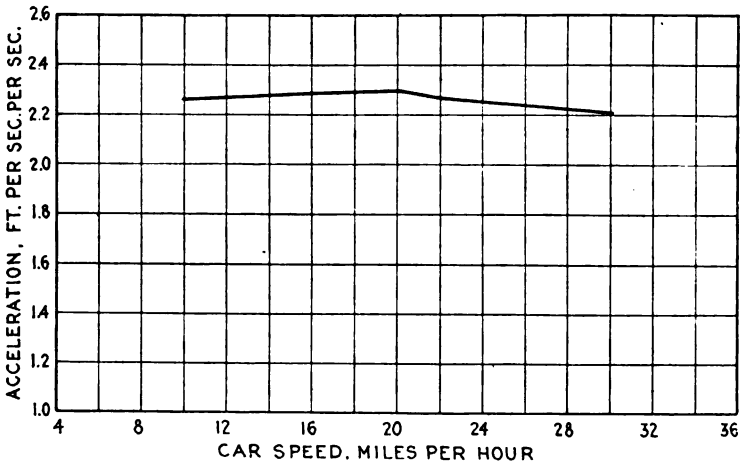

FIG. 3—ACCELERATION CURVE WITH A 3¾ BY 5-IN. FOUR-CYLINDER ENGINE

Axle ratio, 4:1. Gross weight, 3575 lb.; net weight, 3100 lb. Wind, N.W., 10 m.p.h. Top, windshield, curtains up. Road, dry.

The next point in comparing smoothness of operation relates to low-speed running. In this respect the manufacturer of a modern four-cylinder car having a rear-axle gear ratio greater than 4.5 to 1 need have no hesitancy whatever in comparing his product with any car made. Cars having six, eight and twelve-cylinder engines will give a better performance at low speeds when the engine is suitably adjusted and carefully tuned for the test, but it is the almost universal experience of car owners that it is difficult to keep the multi-cylinder engine tuned to the point where it will give a demonstrating low-speed performance. It is only upon the cars whose owners take exceptionally good care of them, or which are driven and cared for by unusually good chauffeurs that the low-speed advantage of many

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cylinders becomes apparent. A great deal has been written regarding over-lapping power impulses of six, eight and twelve-cylinder engines, without taking into consideration the fact that before these impulses are transmitted to the driving mechanism of the car they are absorbed by the flywheel, and that by suitably proportioning the flywheel great smoothness at slow speeds can be obtained with few cylinders.

Going from one extreme of engine performance to the other, let us compare the high-speed possibilities of the different types. It would seem that in this direction engines of a great number of cylinders should be preeminent, but up to the present time at least they do not appear to have proved so. Apparently the most efficient

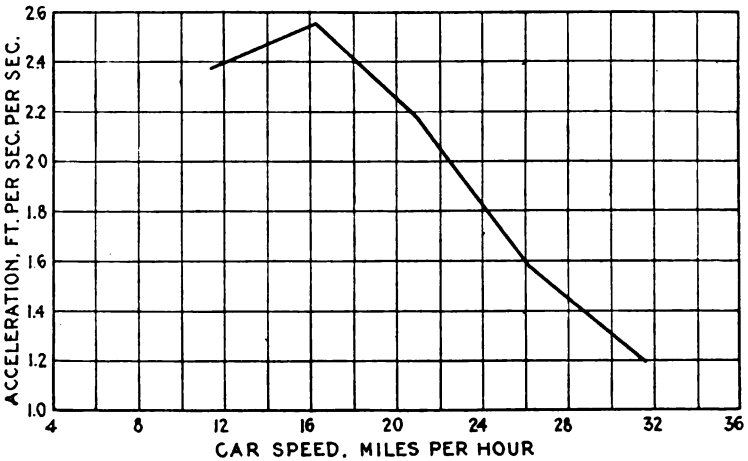

FIG. 4—CURVE OF ACCELERATION WITH 4⅛ BY 4½-IN. FOUR-CYLINDER ENGINE

Axle ratio, 3¾:1. Gross weight, 3300 lb.; net weight, 2875 lb. Wind, N.W., 20 m.p.h. Top, windshield, curtains up. Road, dry.

high-speed engines made are those of four cylinders having four valves per cylinder placed in the head. I have had no personal experience with engines of this type, and can judge only from published records of track performance. In touring car engines, however, the high-speed possibilities of the four appear to be well beyond all ordinary requirements. Reciprocating parts have decreased in weight remarkably within the past few years. Several years ago I found the reciprocating weight to be about 0.4 lb. per sq. in. of piston area. At the present time this has been reduced to about 0.2 lb.

As the formula* for inertia forces due to reciprocating weight is

$$\frac{F}{a} = 0.00034 \frac{W}{a} N^2 r \ (\cos\theta + a_1 \cos 2\theta)$$

the inertia forces of the average engine have been halved for an equal

*For derivation of formula see C. E. Lucke's "Gas Engine Design," page 82.

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number of revolutions. Internal forces due to speed have been reduced proportionately and the possibilities for high speed are apparent.

Improved lubrication also has accomplished wonders, and only recently the matter of counterbalancing crankshafts, which practice has long been followed abroad, has been brought forward prominently in this country. Although the friction can be considerably reduced in complicated forms of crankshafts, such as are used in many of the six and twelve-cylinder engines, but little is accomplished by applying counterweights to four-cylinder crankshafts for ordinary touring car speeds.

Friction tests of four-cylinder engines show no marked improvement below 2500 r.p.m. At very high speeds there is a probability that counterweighted crankshafts show up to better advantage, but

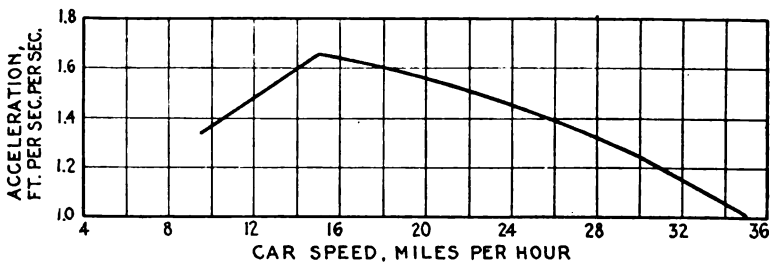

FIG. 5—ACCELERATION CURVE OF EIGHT-CYLINDER 2⅝ BY 5⅛-IN. ENGINE

Gross weight, 4000 lb.; net weight, 3500 lb. Wind, E., 4 m.p.h. Top, windshield, side curtains up. Road, dry.

such high speeds are not encountered in practice on ordinary touring cars.

ENGINE VIBRATIONS IMPORTMENT AT MEDIUM RUNNING SPEEDS

It is, however, at medium speeds that the question whether an engine has objectionable vibrations becomes most important. Motor car vibrations that affect the passengers' comfort can be divided into two classes: First, those caused by the road surface; second, those caused by the engine. The former are of course the most disagreeable, and unfortunately they are also the least controllable. Vibrations due to the engine are largely controllable by the design of the car. It is, I think, almost universally recognized that the vibrations caused by a well-designed four-cylinder engine with light reciprocating parts are negligible at slow speeds. It is not however generally recognized that it is possible to reduce the vibrations at any speed so that they are comparable to the vibrations at slow speeds.

The fundamental formula* for the vibrations of a four-cylinder vertical engine is

$$D = 0.302 \frac{r\,w}{W}$$

where D = displacement of engine from its normal position; r = crank

*For derivation of this formula see *Horseless Age*, Vol. 21, p. 513.

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radius; w = weight of reciprocating parts; and W = total weight of the engine. It will be noted that the speed does not enter into this formula. This is due to the fact that while the forces that cause vibrations increase with the engine's speed, the time in which these forces act in a given direction to produce vibration decreases directly with the speed; hence the decrease of the time factor exactly offsets the increase in the force factor and leaves the tendency for the engine to vibrate the same regardless of speed. This theoretical consideration would be true in practice if all the parts of the engine were perfectly rigid, if it were mounted upon a rigid support, and if the time of vibration of the support did not synchronize with the engine vibra-

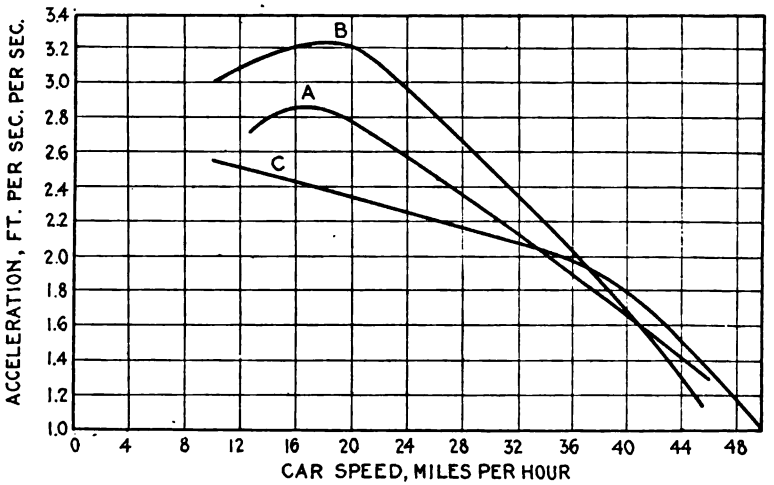

FIG. 6—COMPARATIVE CURVES OF ACCELERATION WITH TWO EIGHTS AND A FOUR

	A	B	C
Engine	8-Cyl., 3 by 4¾ In.	4-Cyl., 3¾ by 5½ In.	8-Cyl., 3⅛ by 5⅛ In.
Gross weight, lb	3,530	3,210	4,470
Net weight, lb	3,090	2,750	4,120
Axle ratio	4.9:1	4.64:1	4.36:1
Top, windshield	Up	Up	Stowed
Curtains	Stowed	Up	Stowed
Road	Dry	Dry	Dry
Wind	Light east	Light east	Light east
Valves	1¼ in.	....	1½ in.

tions at any speed. In practice, none of these assumptions is quite true, but it is possible by making the reciprocating parts light, the crankshaft and other revolving parts rigid, and by suitably proportioning the frame and spring supports, to reduce remarkably the vibrations of a four-cylinder engine.

The synchronous vibrations are not by any means limited to four-cylinder engines. They are likely to be even worse in six and twelve-cylinder engines on account of the more complicated forms of their crankshafts. Sixes, eights and twelves are likely to give

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more trouble from vibrations of the higher periods than are fours, and these vibrations are likely to manifest themselves unexpectedly in parts of the chassis which are distant from the engine.

It is not fair to compare the running of a modern multicylinder engine with that of an old-style four as is so often done. When they are compared with fours of recent design, little difference will be detected.

DURABILITY OF ENGINE, RUNNING GEAR AND BODY

Each part can be made as ample for its work in one type of engine as in another, but the four-cylinder engine should have a longer life than the other types, for it is primarily more accessible and easier to care for, and hence gets better average attention.

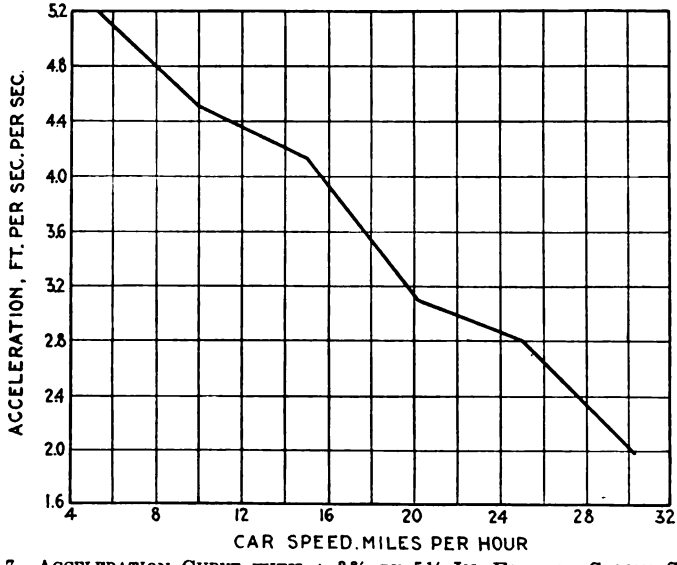

FIG. 7—ACCELERATION CURVE WITH A 3¾ BY 5½-IN. FOUR ON SECOND-SPEED GEAR

Axle ratio, 4:1. Gross weight, 3575 lb.; net weight, 2900 lb. Top, windshield, up. Curtains, stowed. Wind, north. Road, dry.

The efficient life of a good engine is much longer than that of the average chassis on American roads. Ordinarily, springs wear out first, then the body and the other sheet metal parts, then the frame, then the axles, and last of all the engine. The effect of the type of engine on the life of the remainder of the chassis is debatable. The different engine types have not been in use long enough so that we can be sure. We do know that many chassis stresses increase substantially as the square of the car speed, so high speed naturally tends to wear a car out fast. This is the strongest argument for limiting the maximum speed of the car to from 55 to 60 m.p.h. Given equal maximum speeds, road shocks, dirt, and weather effects are the

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all important factors in chassis wear, so far as I can determine. The slightly greater vibration of a good four-cylinder engine does not seem to be a factor worth considering .

The effect of the greater shocks possible to the driving mechanism from the inertia of the heavier flywheel of a four-cylinder engine can be modified to any extent desired by a suitable clutch design. I believe therefore that a car with a four-cylinder engine is as durable as one with an engine having more cylinders, and that the engine is more durable.

MAINTENANCE COST

It is an axiom in all mechanical work that the less complicated a machine is, the lower must be its maintenance cost. Now where

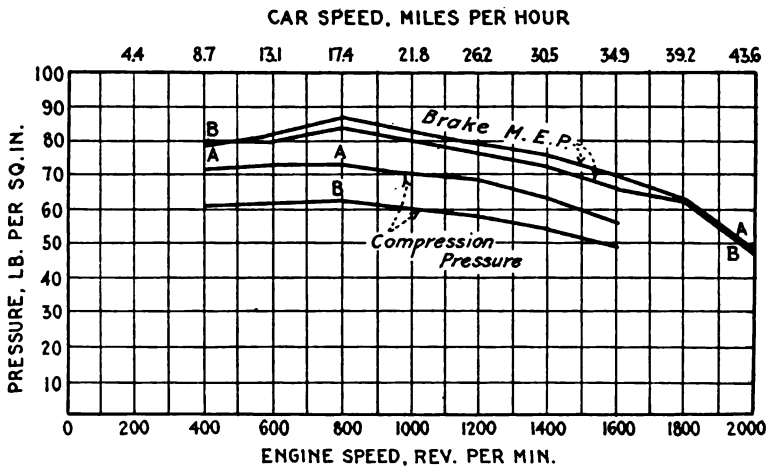

FIG. 8—BRAKE MEAN EFFECTIVE PRESSURES AND COMPRESSIONS FOR TWO PISTONS

Curves *A*, Piston with Length Increased ¼ In. Curves *B*, Standing Piston

is this more apparent than in the comparison of the four-cylinder engine with other types? It is more accessible than other types. It has fewer parts in proportion as it has fewer cylinders. Repairs required are not only less frequent than in other types of engines, but they are less costly to make. There are fewer valves to grind; cylinders to decarbonize; spark-plugs to clean; ignition wires to watch for leaks, breaks, short-circuits and static effects; ignition contacts to synchronize; and less work of all sorts to do. Of course this greatly decreases the maintenance cost to the owner, or the time he has to spend in getting his engine in condition if he does the work himself.

The fuel economy of the four-cylinder engine is fundamentally superior to that of other types. In any type of fuel distribution used in motor cars to-day, the division of the gaseous stream decreases economy. The single cylinder is the most economical type; and the

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fuel economy decreases as the number of cylinders increases. The public is often misled by certified tests made on carefully tuned and adjusted cars, but the result in the hands of the owners is the only thing that will count ultimately. Naturally the rising cost of fuel has resulted in extravagant statements regarding fuel economy, but for the most part these are not borne out in every-day experience.

The designers of multicylinder engines are now at work on much smaller units, utilizing very high compression. This method was tried out about ten years ago, and abandoned, but with the increased technical skill available at the present time, and the smaller cylinder sizes, it is possible that something can be accomplished. It is, however, doubtful just how much can be done with our present motor car fuel, for the gasoline now available tends to partake more and more of the nature of kerosene, and does not lend itself exceptionally well to high compression.

Fig. 8 shows a recent test along these lines. The lower curves marked *A* and *B* are compression curves taken from the same engine using two different sets of pistons. The corresponding upper curves show the brake mean effective pressure obtained from the two tests. It will be noted that it takes a considerable increase in compression to get a slight increase in mean effective pressure, particularly at the lower speeds. It was also found that the engine with high compression did not operate as smoothly as the one with the lower compression. I believe that engines having very high compression, while they may show better fuel economy during tests and in the hands of careful users, will not prove satisfactory for the use of the general public. Possibly, however, other methods of using liquid fuels will solve this problem.

AUTHOR'S CONCLUSION

For cars of the first and second classes (shipping weights not more than 3000 lb.) I do not believe more than four cylinders are desirable. In the hands of the average user a modern four-cylinder car will do anything he requires, as pleasantly as any other type, and will do it more cheaply. This being so, more than four cylinders are not justified.

In cars of the third class (shipping weight more than 3000 lb.) a different condition arises. Here the demand for luxurious appointments and high speeds make the cost of the engine a much smaller percentage of the total cost of the car. The operating cost also is not so important. The weight to be moved is greater, so that here the multicylinder engine has its place. This is its only proper field of automobile application, and I venture to predict that in five years 75 per cent of the money spent for pleasure cars will be for four-cylinder machines.

The buying public is, however, the final judge. Its verdict will not depend on technical arguments but will be based on practical performance. What it will be is not a question for calculation but rather for the application of common sense.

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FOUR-CYLINDER ENGINES OF TO-DAY*

By Finley R. Porter

(Member of the Society)

Abstract

The paper gives the value of certain factors in engine design that are good practice and uses these values to calculate the horsepower of a four-cylinder engine. The author holds that the deciding factor in comparing four-cylinder engines with those of the same displacement but with a greater number of cylinders, is the thermal efficiency. Both the cooling medium and the mechanical losses increase in proportion to the number of cylinders. He suggests in closing, that the demand for power output beyond the possibilities of four cylinders must be met by the use of a greater number.

The few limiting factors of four-cylinder engines, so far as mechanical ability is concerned, are as a rule well understood. The piston areas come nearest to determining the possible power output of a single cylinder. The extreme heat conditions that pistons are subjected to limits the area exposed to the amount that can be kept down to the proper temperature by heat flow to the cylinder walls. While this factor varies according to the mean effective pressure, it has been pretty well established at about 16 sq. in., which represents a bore of 4½ in.

Piston speeds are apparently governed by the ability of bearing metal to withstand the inertia forces. While different materials can be employed to decrease these forces, surface friction becomes a considerable factor at high speeds, so that the practical speed has been established at about 3000 ft. per min. The stroke-bore ratio is purely a matter of choice, but is generally accepted as more conducive to thermal efficiency as it increases; a value of 2 seems about the limit.

Compression pressure close to the point of pre-ignition is had almost universally when extreme power output is needed. While the volumetric efficiency controls somewhat the possible ratio of clearance to displacement, a value of 5 is generally accepted, representing a pressure of about 100 lb. gage.

The mean effective pressure is almost in direct proportion to the volumetric efficiency and is about 40 per cent more than the compression pressure. The brake mean effective pressure varies as the mechanical efficiency and ranges between 20 to 30 per cent more than the compression pressure.

The matter of power impulse per revolution, or torque quality, is not a new subject. Many years of steam practice have established the fact that so far as working ability is concerned, an impulse stroke

*Paper presented at joint meeting of the S. A. E. with New York Section of American Society of Mechanical Engineers.

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every 180 deg., or two per revolution, produces an almost perfect power flow. In the four-cylinder engine there is of course a power stroke every 180 deg.

IDEAL FOUR-CYLINDER ENGINE

Having outlined the factors as we know them to-day let us consider a four-cylinder engine using the specified values, which are not extreme by any means. We can assume the brake mean effective pressure is 120 lb.; the stroke 0.5 ft.; the area of piston 7.06 sq. in., and the number of power impulses 6000 per min. Expressed by the usual formula we have $\frac{120 \times 0.5 \times 7.06 \times 6000}{33000} = 77$ hp., a truly remarkable performance when we consider that such an engine complete should weigh about 400 lb.

This engine applied to the road should be geared about 4 to 1, with a third-speed reduction of 1.33 to 1; second-speed 2 to 1, and first-speed 3 to 1, making the engine-to-wheel gearing 4, 5.32, 8 and 12 to 1 respectively. Using 34-in. wheels, road speeds from 8 to 70 m.p.h. could be had on direct drive, while third speed would afford a possible range from 4 to 65 m.p.h.

The high mean effective pressure obtained would not permit a wide-open throttle at low speeds with any spark advance, as objectionable and detrimental pounding would result. To safeguard against this abuse a purely mechanical carbureter should be employed, with dimensions that prohibit a wide-open condition at low engine speeds.

If greater car speeds are desired, especially for the heavier type of vehicle, more power is easily had by changing the stroke-bore ratio, still keeping within the limits of the assumed mean effective pressure and piston speed. This would, however, result in a slight loss of thermal efficiency.

Constructional features are matters of individual taste. To obtain, however, the efficiency outlined, compression space must be kept to a minimum, which excludes practically all valve types other than overhead or sliding sleeve. Intake gas speeds should be kept below 12,000 ft. per min.; reciprocating parts must be extremely light.

In comparing the four-cylinder engine with those of the same displacement but with a greater number of cylinders, the deciding factor is the thermal efficiency. The heat losses represent practically 60 per cent of the total heat in the fuel used, and are about equally divided between the losses to the cooling medium and to the exhaust. The cooling-medium losses are in proportion to the surface exposed and the heat gradient. Since the surface exposed increases rapidly in relation to the displacement as the cylinder volumes are divided, the heat losses are likewise increased. In addition the mechanical losses, while small, increase in proportion to the number of cylinders. Anything that tends to decrease the thermal efficiency should be con-

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sidered a step backward. The demand however for power output beyond the possibilities of a four-cylinder engine must be met by a greater number of cylinders.

FUTURE DEVELOPMENTS

The future, to my mind, will be full of more interesting developments. So far we have simply spoiled the problem by getting an answer quickly, regardless of its correctness or cost, and I believe all will agree that the cost has been high.

The question of fuel cost cannot be ignored. Increasing the thermal efficiency is the real problem that demands an answer, not so much as a matter of cost per horsepower as of cost per mile traveled. In order to accomplish any results worth while, the cycle must be changed. The induction stroke must be shortened in relation to the expansion stroke; the initial pressures must be materially increased, and the final pressures decreased.

C. E. Sargent in a paper before the Indiana Section of the Society sounded the keynote of this advance. While I do not agree with the method he suggests, the principle aimed at I believe to be correct. (This paper is given elsewhere in this volume of the TRANSACTIONS.)

Working temperatures will have to be raised also, and for aeroplane engines I believe direct air-cooling will play an important part in the development.

These problems are going to prove much harder than past developments. To facilitate an answer being had in a reasonable time, I suggest that the engineering societies promote discussions and comparisons of experimental results. This would result in combined effort and in development worth while.

DISCUSSION OF PAPERS ON FOUR-CYLINDER ENGINES

AERIAL FIELD

F. W. BARKER:—The aeroplane must have a multicylinder engine and should have the particular attention of engineers. The development in the automobile engine has reached a high point. The aerial field now offers ample scope for the wonderful skill developed in the past twenty years by the profound study that has been made of the automobile engine.

H. C. GIBSON:—I take exception to Mr. Barker's statement that the multicylinder engine is necessary for aeronautic purposes. Since economy is perhaps the prime necessity in aeroplane engines, after power, of course, the fewer cylinders the better. Advocates of the greater number of cylinders confuse the advantages of smoothness and absence of vibration with the necessity for economy and large amount of power from a small amount of weight, which are the real necessities in aeronautic engines. We have nearly gone the limit, if Mr. Porter is right in saying that the maximum cylinder bore is 4½ in.

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F. R. PORTER:—From my experiments the 4½-in. piston seems to be the limit. If a much larger piston is used, the mean effective pressures cannot be increased. The mean temperature is almost proportional to the mean effective pressure. I have used steel pistons with a mean effective pressure of about 149 lb. and melted them so that the head would drop out completely. As the piston is made larger, the pressure must be lowered and nothing is gained.

L. G. NILSON:—We must have multicylinder aeronautic engines because when the dimensions are increased the weight goes up faster than the power. The single-cylinder engine may be theoretically more efficient than the multicylinder one of the same horsepower, but the thicker cylinder wall, supporting structure, etc., increase the weight to such an extent that we can produce a lighter engine by using a greater number of cylinders.

FLYWHEEL ACTION OF PROPELLER

F. J. ARMSTRONG:—The propeller plays an important part in the application of engines to aeroplanes. An engine may run well alone, but the propeller introduces various forms of skin friction, which causes very peculiar effects.

C. W. HOWELL, JR.:—In the aeroplane, we discard the flywheel to save weight. The propeller must act more or less as a flywheel. The propeller is neither spherical nor round, is therefore difficult to balance and subjects the aeroplane to intense vibration. The larger the number of impulses given the propeller the less the vibration and the smoother the running action of the aeroplane.

PROF. F. DER. FURMAN:—The propeller is said to cause vibrations in the Gnome engine. In studying vibration we ought to take into account other factors. If the cylinders are revolving uniformly at 1200 r.p.m. about the center, the pistons and connecting-rods are turning at a variable rate of speed, each revolution, of roughly between 800 and 1600 r.p.m. about their center. The acceleration and retardation of piston and connecting-rod due to this variation can cause vibrations. The lengths of connecting-rods are also extremely important, because if they are not made of proper lengths, the compression will be unequal; with unequal compression vibrations that are not due to the propeller can take place.

H. C. GIBSON:—The propeller, whether it has two or four blades, or even if it is shrouded, is an efficient flywheel because of the speed at which it runs.

C. W. HOWELL, JR:—The most unstable medium we have to work in is air. It is never constant; it is never the same any two moments. When the propeller is affected each instant by a different pressure, it cannot act efficiently as a flywheel.

INCREASING EFFICIENCY OF FOUR-CYLINDER ENGINES

E. FAVARY:—Both the papers show that the four-cylinder engine has a higher thermal efficiency and is, therefore, the engine that

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should be most extensively used. An engine with a greater number of cylinders, has, on the other hand, a more uniform torque and a better running balance; it allows driving at low speeds without jerks and such frequent gear changes. All vibrations can be decreased better in engines of a larger number of cylinders than in four-cylinder engines. On account of the increased cost of fuel, economy is most important, and therefore the greatest field for the future, as Mr. Porter pointed out, is to increase the thermal efficiency.

The greater the number of cylinders, the greater the jacket area exposed to the hot gases, and the greater are the heat losses. For instance, a six-cylinder has approximately 20 per cent more area exposed to the cooling water than a four-cylinder engine, and in practice uses somewhere between 10 and 15 per cent more fuel. However, even in a four-cylinder engine the losses are enormous, and I want to point out some means that may prove a basis for future development.

Since piston speeds as high as 3000 ft. per min. are now practicable, whereas some years ago 1000 ft. was considered the limit, the combustion chamber can be kept at a much higher temperature without danger of pre-ignition. But oil will carbonize at higher temperatures, thus rendering its lubricating qualities defective and the running of the engine impracticable. A water-jacket might be provided for only the part of the cylinder below the combustion chamber, so as to cool the portion requiring lubrication, while at the same time allowing the combustion chamber to have a higher temperature. The cylinder head could be bolted to the cylinder and asbestos packing or some other non-conductor placed between them. To get the best results the piston-rings should be lower down on the piston so as to increase the portion of the cylinder that could be maintained at a higher temperature. The temperature of the combustion-chamber wall might be still further reduced by lining it with some material substantially impervious to heat.

A higher engine speed, as previously mentioned, would greatly assist in overcoming pre-ignition. In order to keep the mixture cool, it might be advantageous to compress it outside the cylinder, and admit it through large valves into the cylinder, after the completion of a portion of the suction stroke. The mixture, at a low temperature, entering the cylinder under compression, requires some time before reaching the temperature of spontaneous ignition.

Furthermore, in forcing the compressed mixture into the cylinder at a certain time of the suction stroke, the gas will expand, thereby tending to lower the mixture temperature. On the compression stroke, of course, the temperature will again be raised. This, with the increased engine speed, might make it possible to have the combustion chamber at a much higher temperature, thereby obtaining an increased thermal efficiency.

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COMPRESSION AND MEAN EFFECTIVE PRESSURES

A MEMBER:—Why does not Mr. Porter agree with the method suggested by Mr. Sargent for increasing thermal efficiency?

F. R. PORTER:—Mr. Sargent advocates an adjustable admission-cam, which permits increasing the compression pressure at higher engine speeds. The idea was to control, not by throttle but by admission of gases, the throttle being wide-open practically all the time, cutting off earlier at low speeds and increasing the length of admission stroke when more power is desired. The plan has many advantages but would probably add a number of complications.

I have worked along the same line of thought. I have a touring car in operation with some radical features. It is a seven-passenger car with a 140-in. wheelbase, weighing 3470 lb. The engine has a 4.6-in. piston and its compression is 115 lb. If the throttle were opened up at ordinary engine speed, a cylinder head would be knocked off in a minute. I never use a wide-open throttle, but the mean effective pressure is high all the time; even when climbing the steepest hill the throttle is not over one-eighth open. The throttle would be wide open only at extreme speeds, say at 95 m.p.h.; but the effect on intermediate speeds is wonderful. The car is geared 3 to 1 with 36-in. wheels; it will perform satisfactorily above 4 m.p.h. on direct or any test we have made. I get 12½ to 16 miles to the gallon with the car, but this is due entirely to the fact that I am carrying a mean effective pressure of 60 to 80 lb. at one-fifth throttle. It was an accident in my case, but Mr. Sargent advocates this high mean effective pressure at lower speeds.

TEMPERATURE LIMIT IN STATIONARY GAS ENGINES

J. H. NORRIS:—With the fuel now obtainable it is not safe to run stationary gas engines with the combustion chambers at higher temperatures than are now being used. If the temperature of the combustion chamber or of compression is carried too high, liquid fuel will crack on entering the cylinder and ignite before the proper ignition point, resulting in severe gas knocks. Experience with engines having cylinders from 3 to 18-in. diameter has proved to my satisfaction that development for greater economy will have to follow other lines than those of higher temperature and greater compression.

F. R. PORTER:—I have found no trouble in using high temperatures but there is difficulty in getting the useful heat out of the large volume of gas before it escapes to the cooling medium. In direct air-cooling the amount of fuel must be kept down to a certain limit, governed by the temperature. There is a gain in thermal efficiency but a loss in power. If high temperatures are made possible by insulating the cylinder heads, the amount of work must be decreased, or the temperature will increase beyond the limit.

Considerable stress has been laid on high compression stationary gas engines. The subject has been dealt with on the theory that the

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temperature of the engine varies as the pressures. But the mean temperature, that is, the controlling temperature, apparently has nothing to do with compression except insofar as the volumetric efficiency and final temperatures enter into the question.

We are experimenting with high compressions, taking the work from the gas before it goes to the cooling medium. We intend cooling it by expansion. But in doing that we are starting with a 4 by 16-in. engine. We really have two 4 by 8-in. engines head to head, pistons together; we will have a 4 by 6-in. induction stroke and a 4 by 8-in. expansion stroke. If the single-cylinder stationary engines had a good compression ratio and were using any great amount of fuel they could not carry so much compression. The temperature would increase tremendously unless the ratio of expansion was very large.

DETERMINING NUMBER OF CYLINDERS

J. EDWARD SCHIPPER:—The number of cylinders seems to depend almost on the piston displacement per cylinder. The best working stress on main and connecting-rod bearings is an important factor affecting the number of cylinders. A piston displacement of about 70 cu. in. per cylinder is close to that used for a number of the satisfactory present-day automobile engines. Such a displacement gives an easy working engine, as far as bearing surfaces are concerned, and also one with good carburetion.

For cars of what Mr. Watts calls the luxurious class, in the case of which he says multicylinder engines may be justified, the piston displacement of 70 in. per cylinder would probably be rather high.

F. R. PORTER:—The displacement Mr. Schipper mentions is a little less than the actual value, but is well within what is necessary. No automobile engine built for any use outside of racing needs more than 300 cu. in. displacement. A 300-cu. in. engine will easily give 100 hp., which the average automobile never needs. At 40 m.p.h. the five-passenger, 2000 to 3000-lb. car takes only about 20 hp.

RELATION OF SPRUNG WEIGHT TO TIRE WEAR

A. E. POTTER:—What is the relative tire wear in cars of the same horsepower, driven by four, six, eight and twelve-cylinder engines?

F. R. PORTER:—I believe that the engine has not much to do with that. The relation of sprung to unsprung weight has a good deal to do with tire wear. In racing or testing work we can go out on an ordinary macadam road in the morning, when it is covered with dew, and if there has not been much traffic see that the car touches the ground only at intervals of from 10 to 40 ft. The jumps increase as the proportion of unsprung to sprung weight increases. We do not think of having the unsprung weight more than 35 per cent of the sprung weight above it. The sprung weight is the fly-

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wheel of the car, in just the same way that the flywheel smooths out the impulses of the engine.

COMPARISON OF AUTOMOBILE AND STATIONARY ENGINES

W. T. PRICE:—The work of the members of the American Society of Mechanical Engineers present is as a rule concerned with larger sizes of internal combustion engines. The cylinder diameter of the automobile engine is said to be limited to 4½ in. Cylinders of about 20-in. diameter, giving each about 150 hp., seem to be about the limit of size in large four-cycle oil-engines, unless water-cooling is used. The single-cylinder engine is, of course, the simplest. Almost anybody can take care of it; ordinarily it requires very little attention. But single-cylinder engines in large sizes are more expensive than multi-cylinder engines.

In the two-cycle internal-combustion engines, the heads and pistons are more likely to crack, and the lubrication problem becomes more complicated. There was a great tendency toward two-cycle oil engines about five years ago, but that has subsided now in favor of the multi-cylinder four-cycle engine.

The multicylinder engine is favored for stationary work because it gives a more uniform rotation, creating a steadier voltage for lighting, without such a tremendously heavy flywheel. In the parallel operation of alternators, the four-cylinder engine is superior to the one with fewer cylinders.

The weight per horsepower of the automobile engine is down to perhaps a minimum of 7 lb. The weight per horsepower of single-cylinder stationary engines is as high as 500 lb. In the four-cylinder engine the weight per horsepower is about 200 lb. The weight of highly developed engines for submarine work is below that but we have never reached any such point of development as has the automobile engineer.

The compression pressure of the automobile engine is limited by pre-ignition to about 125 lb., while a stroke-bore ratio of about 2 seems about the limit. With oil engines, however, we do not have pre-ignition, because combustion does not occur until near the end of the compression stroke. A heavy oil engine runs up to about 300 lb. compression on the Otto cycle and up to 600 lb. on the Diesel cycle. The stroke-bore ratio on large engines has been about 1.8, but there is a tendency to decrease the ratio, keeping up the speed in proportion. The ratio seems to be about 1.2 in European practice.

We do not dare go above 900 ft. per min. piston speed. A stationary engine is built to operate under full load twenty-four hours a day for twenty years, and a different kind of engine is required. I look forward to the time when the stationary engine will be developed somewhere near as much as is the automobile engine. Then we will have a type of prime mover to be used throughout the whole world.

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DIFFICULTIES WITH EXTREMELY HIGH ENGINE SPEEDS

JOHN E. BULLARD:—Why can we not decrease the weight as much by increasing the speed as by increasing the number of cylinders?

F. R. PORTER:—Only the inertia forces limit the speed of the four. We ran one up to 5000 r.p.m., when a connecting-rod broke. All that was left of the engine was one cylinder, the timing-gears and the flywheel. A piston small enough to run at such extreme speed would be practically working against itself, because the heat will not flow from its center. A certain amount of material is needed in the piston; otherwise the heat flow never reaches the walls. No matter what the cooling medium may be, with a certain area and mean effective pressure, the heat does not flow away from the piston and a spot about as big as a dollar will melt in the center. The rest of the piston will not show any undue heat and its lubrication will be satisfactory.

I am experimenting now to find the limit of aeronautic-engine speed that I can maintain indefinitely. We have set nine-hour runs as the test; the engines to come through in perfect condition, without need of repair. We began at 1200 r.p.m., and with throttle wide-open ran nine days for nine hours at 1200 and for the same time at 1400 and 1600 r.p.m.; at 1800 r.p.m. a connecting-rod bearing failed after the first hour. That was to be expected because I had taken an old Knight engine and placed other cylinders on it, so that the bearings were not big enough for the displacement.

We do not figure so much on the number of impulses per revolution of the engine as on the number of impulses per revolution of the propeller. Similarly I cannot understand any reason for gearing a four-cylinder engine four to one and then gearing an eight-cylinder the same. I think the final measure is at the road wheels, so far as power impulses go.

PROPER COMPRESSION IN FOUR-CYLINDER ENGINES

H. C. GIBSON:—Perhaps we are not so satisfied with the four-cylinder engine to-day, as we seem to think we are with the multicylinder, because in the earlier racing engines the compression was as high as it possibly could be in order to get the greatest acceleration and speed. Such an engine is not the kind we want in a touring car. That brings up the point: What compression should be used for a four-cylinder engine?

The compression of a four-cylinder engine must be less than that of a multicylinder engine of the same power, because so much less heat is lost to the cylinder walls in the larger cylinders of the former. The heat not lost to the walls is retained in the mass of the gas, the center of which becomes extremely hot and induces pre-ignition. Therefore we cannot compare the two on the basis of the same compression. The compression in the four must be lower; then the engine will work much more smoothly.

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The four is at least as efficient theoretically, if not more efficient; and is just as efficient practically because less heat is lost to the cylinder walls. The absence of a successful light-metal piston has restricted the use of the four. But to-day we really have light-metal pistons that do not crumble on top, give way and leave the piston without a crown.

F. R. PORTER:—The compression of four-cylinder engines on touring cars is usually entirely too low. I believe compressions of 70 to 80 lb. are general practice. Not 5 per cent of the time is the engine running with a wide-open throttle. That means 30 to 40 lb. compression for average running, which is very uneconomical. Increase that 50 per cent and bring the half-throttle and quarter-throttle position to 60 or 70 lb. compression. The high mean effective pressure is then used on high speed only, where it can be handled. Of course that may be objectionable, particularly to people who will not change gears.

GEAR-CHANGING ADVOCATED

My car has four speeds and I wish it had five. That would add a little too much complication. There is no comparison between a three and a four-speed gearbox. I can come to New York, 64 miles, in an hour and twenty minutes, and not hurry, with a four-speed gearbox. I think the difference would be noticeable with a three-speed in 60 miles, because the car cannot be operated under economical conditions or conditions favorable to the engine.

I change gears at every corner because I like to operate the engine in a way that it will work at its best. It is surprising what a mean speed you can keep up without going over 50 m.p.h. But changing with the usual big reduction between direct and second in a three-speed gearbox is so awkward that nobody ever goes into second, because it means slowing down so in the road speed, which is objectionable. But with a third just a little under high, say 1.33 to 1, there is no objection to running on it. My car is faster than the average, but I can travel at an average speed that is unbelievable, mainly because of the fourth speed.

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MID-WEST SECTION PAPER

ART AND THE AUTOMOBILE

By William B. Stout

(Member of the Society)

Abstract

The automobile of to-day has been developed mechanically to such a high point that the art of building them, according to the author, is gradually becoming a studio task. The paper outlines the rules of appeal in body design and shows how former types of bodies have lost their appeal because they were not based on correct principles. The author gives the ways in which power, body width, mass, comfort and other car qualities can be suggested by proportions of the body or by its color. He states that the automobile of the future will take every advantage of art knowledge to build up an appeal consistent with its mechanical performance.

Art is the science of eye-appeal; the appearance-basis of attractiveness. If one build into a commercial product an appeal to the eye, he establishes the first point of salesmanship, which is impression.

The designing of a motor car is no longer a task for the engineer alone. There was a day when it was merely a mechanism for traveling from place to place, a machine on which seats were mounted that one might enjoy the sensation of swift travel and novelty of mechanical progress. To-day the sensation is old and the novelty become the usual.

With the world's acceptance of the motor vehicle and the perfection of car machinery, a new thing was needed to carry appeal further than mere mechanical construction. Comfort and performance stunts have been made a basis of sales. Freak demonstrations and misleading economy tests were tried and succeeded in attracting. Only recently has the value of eye-appeal been recognized, and art even considered in relation to motor vehicles.

Reliability is the easiest thing to design into the present-day automobile. Any modern motor car will travel from New York to San Francisco with a satisfactory degree of reliability, and some of the cheap cars with greater mileage per day than the heavy weight, more expensive constructions. The difference then between one car and another, after a certain extremely cheap class is passed, is in passenger comfort and the self-respect of ownership. This self-respect or pride-value depends upon the authority of the design, and the degree of art involved in its make-up. The art of car building is thus resolving itself more and more into a studio task.

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If a car is designed for a certain excellence or standard of mechanical performance, its body lines must be so disposed as to proclaim and suggest that performance. If a car is designed primarily for comfort, the art lines should suggest comfort.

BODIES SHOULD EXPRESS INDIVIDUALITY

There are certain definite rules and principles of art which have been applied rarely in car design. They were not made use of until a few years ago. Even yet few companies employ artists on their engineering staff. The day will come when bodies will be designed by artists of national reputation spending all their time to make the bodies express by their lines, contours, and arrangement, the individuality and performance the car possesses. If the body backs up in its appeal the statements of the advertising, and the performance and life of the car back up the appearance, the car will be a success, and the marketing of it accomplished along lines of least resistance and cost. Many a firm to-day, spending thousands of dollars a year in advertising a body design on a good chassis, might save much by spending a small amount on the hire of an artist experienced in body work. There is real commercial value in an unique design of car. This has been lost sight of by many manufacturers. It is the advertising value of every car running.

All cars look alike to most people, merely because the makers have followed each other and made the cars look alike. When one car or another goes by, the public does not distinguish it from the rest. One might hang a sign from end to end to proclaim the make, but the owner would not allow it. If the design be original so that even the man in the street names it as it passes, the car itself is a sign as long as its wheelbase and as insistent as its appearance.

RULES OF APPEAL IN BODY DESIGN

The rules of appeal in body design are not intricate but are easily understood. All of the principles are, of course, not treated in this paper, nor can their application to individual requirements be detailed for different cars. The main ideas can, however, be outlined. One generally speaks of the "lines" of a car. It is true a car has lines but they are not the greatest attraction-basis of design. We speak of the human figure or face as having lines, but the profile of the face, for example, is not the most appealing view. Too often the profile of a straight side-view of a car is laid out on the board quarter or half-size and the art "lines" judged from this view. This is a view never had by the passer-by and if it were possible to have it there would be no life or action in it, and hence no appeal. All appeal to the beauty sense is by suggestion or implication. Mere lines cannot suggest to nearly as great an extent as masses and edges of shadows can. With the shadows should be included the reflections, which furnish new suggestive edges leading on the subconscious reasoning which gives us our final appeal-decision. This is why a newly

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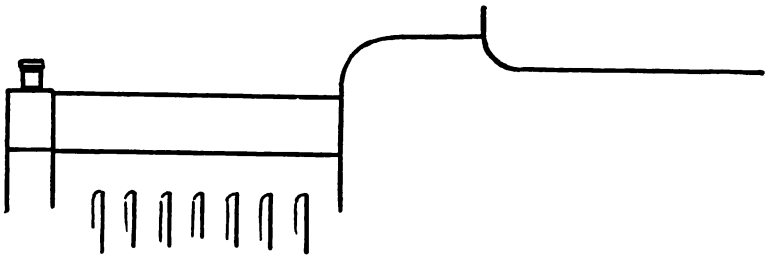

FIG. 1—HOOD AND COWL WITH BREAK BETWEEN THEM

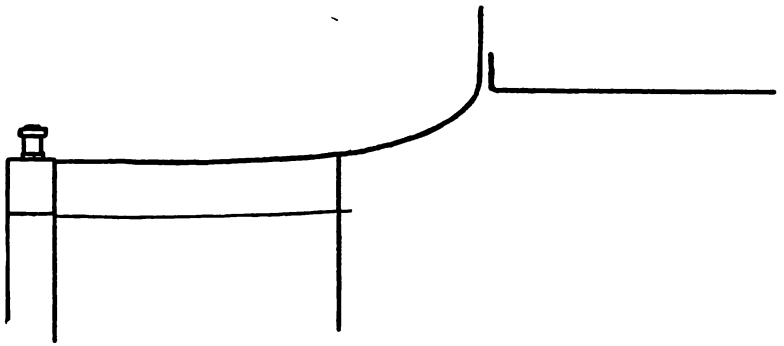

FIG. 2—COWL AND BODY LINES GROWING OUT OF HOOD LINES

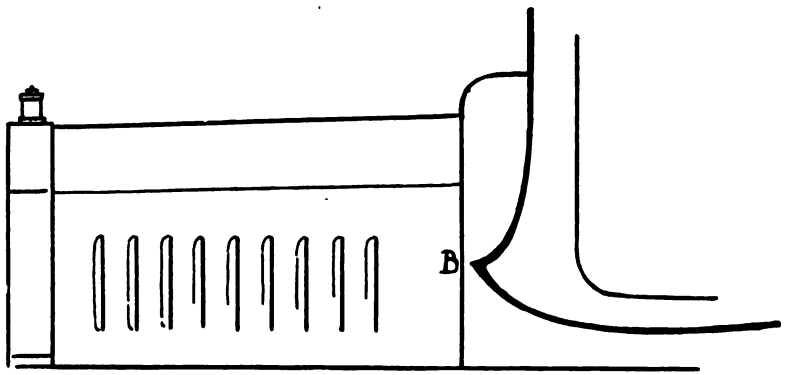

FIG. 3—POINTED DOOR OR BODY HOOK JUST BACK OF HOOD

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finished, highly polished body presents such an extreme appeal, compared with the same car with a dull finish, unless the car has been designed primarily for dull finish by having contours which make up for the lack of reflection-suggestion.

Types of bodies built a few years ago have lost their appeal because they were not based on correct principles of art. In nature one thing grows from another, like the branches from the tree trunk, and the twigs from the branches. The newer stream-line bodies aim at this principle in having lines continuous from front to rear with no break, one growing out of the other, but these are pleasing only to the extent that other principles as well are followed. Fig. 1 shows a hood and cowl with a definite break between these parts—a reverse curve which, to look right, would necessitate a hood of different color from that of the cowl and body, to give a reason for the break.

The sketch in Fig. 2 shows the cowl and body lines growing out of the hood lines, as introduced first in this country by the Hudson, I believe, this being one art reason for the great appeal of its body design. The limousine top in this make of car did not look like an attachment or a box set on a touring car, but, by being continuous in line with the lower design, as part and parcel of and integral with the lower section. This idea of line continuity is one of the greatest problems in the stream-line curved-top bodies.

SUGGESTIVE LINES

We have learned, however, that shadows and reflections are more important than mere lines, and that one must grow from another or from a focusing curve. It is admitted generally that horizontal lines suggest speed and vertical lines stability. This is not true unless the eye hit the right part of the design first. To suggest speed the eye must first alight on the front end of the car and then travel back. Horizontal lines, once the eye is sent to the front end, will help to carry it back, and for this purpose are necessary, but they are useless unless the eye starts at the right place. This illustrates the reason for the nickeled radiator or unusual fender arrangements in front on several cars. The Oakland has used the prominent radiator feature for some years, while the Paige in the newer models gains front-end attention by an unusual front shape and fender splash. Packard and Hudson gain attention by unusual radiator caps; the result is the same.

Vertical lines on a car indicate stability only if the eye travels upward from the bottom. One of the reasons for the charm of the pointed door or body hook just back of the hood, as used on coupé and limousine models of prominent makes, is that this accent, shown in Fig. 3 at *B*, draws the eye to the bottom of the door first and then allows it to travel up and back. A curve to the roof, leading up and and back also adds to the speed feeling and appeal of the design. A car with a flat roof and a big ventilator or railing on top looks unstable, rather than sturdy, even though the vertical lines are made more insistent.

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IMPRESSIONS FROM PROPORTIONS

The sense of power in a car can be suggested in two ways; first by the proportion of hood length to the rest of the car, and second by hood height compared to body-sill height.

Body width gives the impression and fact of roominess, while height from the ground suggests stability. Length gives the idea of ability to hold the road at speed, provided the mass of the body's design is carried at the front.

Many cars fail in appeal by being too high in proportion to their width. This is sometimes due to frame necessities when a straight frame is used, and sometimes to mechanism that is carried too high. Cubical dimensions of the body give one a sense of weight or lightness in relation to the car, and by suggestion of operating expense appeal or repel. To look roomy a car must appear wide in proportion to its depth.

The mass of the running gear, the heavy look of the wheels, etc., suggest comfort. Big wire wheels of red or white denote luxurious riding, though they may ride no more easily.

A great deal of the pleasure of riding is marred by uncomfortable seats. Luxury in cushioning is to be sought, and high curved backs deeply tufted add to comfort, as well as appeal to the eye.

EFFECT OF COLORS

The final appeal of exterior design is obtained by color and finish and this will depend largely on the experience of the buyer. Black brings out lines and reflections and hides shadows. White emphasizes mass and kills color. Strong colors do not appear well in combination with white, except in small spots or accents, or in striping. Black is a good color for a car of beautiful outline. White is preferable for a car of mediocre outline and gives a sense of largeness. This applies also to tones and colors between white and black. There is a color which best fits a given car. Manufacturing reasons, however, hinder the search for this ideal.

Eventually color will be chosen by the buyer for utilitarian reasons. A color that shows dust and dirt least will be preferred. This points to lighter colors, as black is hardest of all to keep clean, though the easiest to put on in manufacture.

A small vehicle needs more careful finish and appointment than a large one to obtain its measure of road respect. A big limousine obtains its road authority through its mass. A small car must obtain this through insistent appearance. Better finish and trimming almost to the point of flashiness are needed in a light car. What would be bad taste or poor art on a big vehicle becomes good taste on a light car for this reason. Enameled leather upholstery and a quilted dash might look out of place on a big car, but they are quite the thing on small construction.

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CONCLUSION

The car of the future will be designed with art lines to suggest the action of its mechanism. It will take every advantage of art knowledge to build up an appeal consistent with its mechanical performance. Speed, power, comfort, luxury, safety and economy will be suggested by appearance, by following art rules as faithfully as engineering rules are followed in the shop.

The automobile for to-morrow is the artist's opportunity.

DISCUSSION

G. W. SMITH:—I have been somewhat of the same opinion Mr. Stout is, that the artist who designs the lines and curves of a car has as much to do with marketing it as has the engineer. It is obvious that the car is designed to express a form of beauty rather than a thing for use.

HENRY FARRINGTON:—I was in an art department one day when the artist was looking over some pictures of a certain touring-car model. He added a few lines and rounded off a few corners. The actual lines of the car had not been altered by more than an inch in any one place, but with a few pencil strokes he had outlined a car that possessed the eye-appeal mentioned in this paper. The maker decided to build a body according to the sketch, and I am positive it will be a great success.

PORTER E. STONE:—The firm with which I am connected builds bodies for custom trade only. We are coming back again to the brougham or closed vehicle for town use. The town vehicle demands a different type of body from that of the factory vehicle being turned out to-day. The class of varnish is changing for the road vehicle; the varnish for the town vehicle will probably always have high polish or finish. In 1834 we had no varnish of any sort.

We find in the body business three or four divisions of art. The Germans have one type, the French another, and the English another. In the first automobile we seemed to follow the English design, that is, curves and serpentine lines. Next came the French vogue with its melting and artistic bodies. Suddenly we have turned to the streamline or German type.

WILLIAM B. STOUT:—These remarks bear out rather than contradict the idea I was trying to express as to the lines suggesting performance. In the custom-built body the lines are made to express something else than the performance of the car. In the highest class of work we are trying to get away from an instrument of color. We are trying, I think, especially in the town car at present, to express class and refinement.

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